The
ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

JUL 30 1942

VOL. III. No. 5 MAY, 1942

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Published Monthly at 120, Wigmore Street, London, W. 1.

Publishers : **ENGINEERS' DIGEST LTD.** - - - - Telephone : WELBECK 9357

Advertisement Dept. - 120, Wigmore Street, London, W.1 - Telephone : Welbeck 9357

Distribution - - E. & F. N. SPON, LTD., 57, Haymarket, London, S.W.1
Telephone : Whitehall 1860. Telegrams : Fenspon, Lesquare, London

Editorial Dept. - - 120, Wigmore Street, London, W.1 - Telephone : Welbeck 9357

Subscriptions to : ENGINEERS' DIGEST LTD., 120, Wigmore Street, London, W.1, or
E. & F. N. SPON, LTD., 57, Haymarket, London, S.W.1

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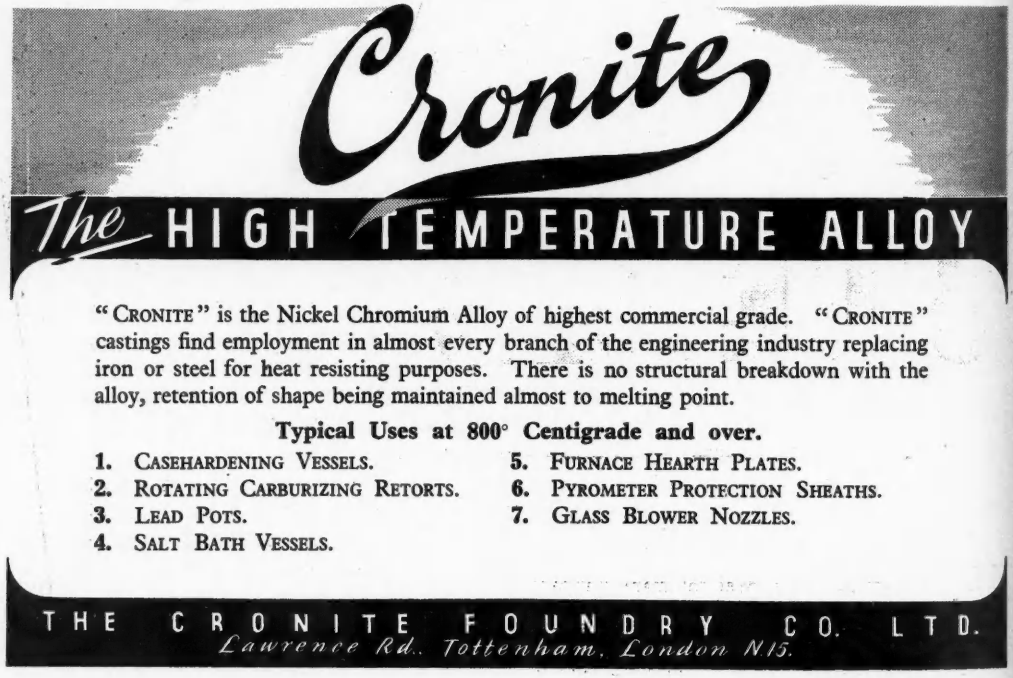
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CONTENTS

AERONAUTICS : *page*

Wind Tunnels . . . Birthplace of Streamlining 165
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 165.

Power for Man-Made Hurricanes 167
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 167.

Stressed Skin Structures in Flywood 187
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 187.

ELECTRICAL ENGINEERING :

An Automatically Reclosing Breaker for Co-Ordinating Distribution Systems 182
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 182.

Radiography at High Speed 185
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 185.

Electric Traction Vehicles 187
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 187.

Electric Elevators in Industrial Premises with Explosive Atmospheres .. 191
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 191.

MECHANICAL ENGINEERING :

Utilization of Heat in Industrial Furnace Plants 162
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 162.

The Influence of Compressibility of the Fluid on the Properties of a Centrifugal Blower 170
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 170.

The Separation of Impeller and Diffusor Losses in Radial Blowers .. 174
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 174.

The Determination of the Variation of State in Turbo-Machinery by Means of the Increase in Entropy 178
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 178.

Saving Fuel in Gas-Heated Furnace Plants 195
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 195.

METALLURGY :

Low-Tin Solders 164
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 164.

The Use of Pressed Metal Powders 194
" *The Engineers' Digest*," *Vol.* 3, *No.* 5, *p.* 194.

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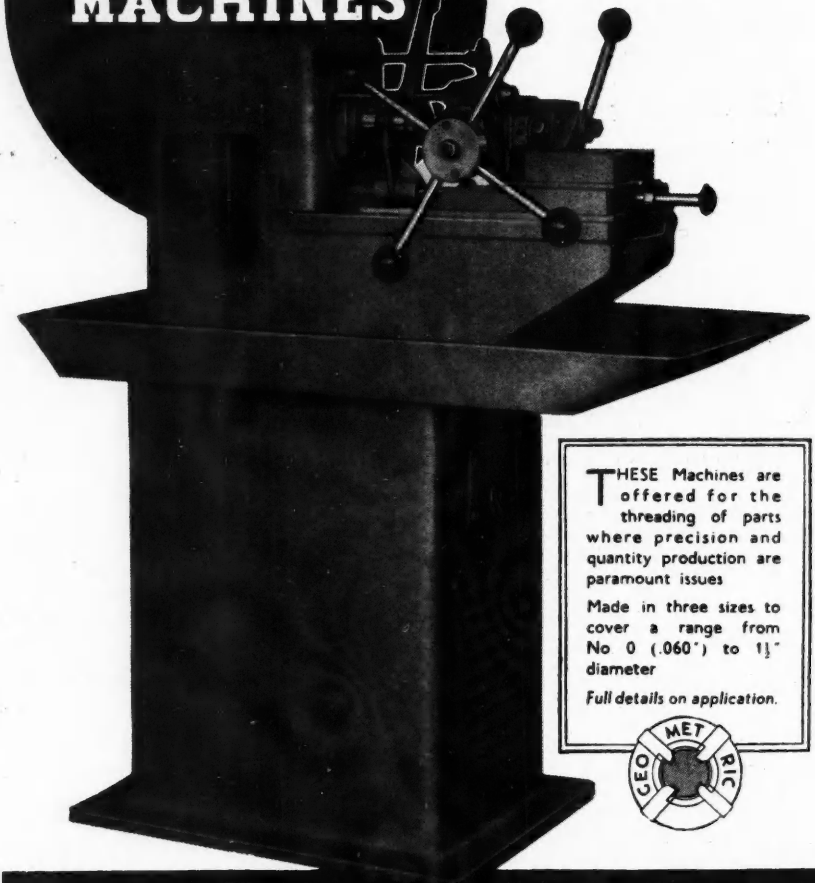

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EDITORIAL

FUEL ECONOMY

THERE are few problems which have received greater publicity recently in the daily newspapers than the Government's plan for domestic fuel rationing. Our technical journals, however, have been paying much too little attention to this question though it is, essentially, a problem for engineers to solve.

If we take it as a dual problem of increasing coal production and saving on the industrial consumption of solid fuels—and it is our opinion that this would have been the right way to tackle the question—it will be obvious at once that engineers alone could find the methods and means to deal with the matter adequately.

Our present coal production amounts to 205,000,000 *tons per year against* 280,000,000 *tons before the war, from which latter figure, however,* 50,000,000 *tons per year were being sent abroad as exports. The requirement of the industry is an additional* 10,000,000 *tons or about* 5% *of to-day's consumption, which the Government wanted to save by rationing domestic fuel consumption, which must result in colder and darker homes for the whole population.*

There has been no grumbling about the rationing of food, petrol and a number of other more or less essential commodities in this country. The British people have shown that they would willingly endure any sacrifice necessary to achieve victory.

It can be shown that rationing of the domestic fuel consumption is not necessary. There are certain alternative methods of meeting the fuel situation. It can be stated without exaggeration that, by exploiting possibilities resulting from efficient organization and by the application of mechanical means, the coal production can be improved upon by at least 2 *per cent (about* 4,000,000 *tons per year) without additional mining labour. The concentration of the available labour on the best seams in the best mines, suitable measures to prevent absenteeism, and the improved exploitation of the outcrop seams are some possibilities on the organization side, while the technical improvement of production can be entrusted and will be solved by mining engineers in every district according to the peculiarities of every pit.*

Even when neglecting the additional quantity resulting from improved coal production, the 10,000,000 *tons of coal required can be saved by the industry itself. It must be an approximately correct estimate that the requirement of the industry, including the consumption of gas and electricity undertakings absorbed by the industry and that of the railroads, is about* 60% *of the total needs of the country. Thus a saving of* 10% *in the industrial consumption would offer an ample solution of the problem.*

Economies in methods of combustion, preheating air, better use of blast furnace gas, coke-oven gas and other fuels resulting from the use of coal in the iron and steel industry, the more efficient use of steam through superheat and higher pressures, the extensive adoption of stokers on the railroads, feedwater treatment eliminating tube losses and the formation of scale from impure water, and a great variety of progressive methods and devices for the manufacturing industries are only some of the subject matters of a series of articles which will be published in this and future issues of THE ENGINEERS' DIGEST, *showing clearly methods of obtaining the necessary coal for the war industries by saving, without the necessity of harming the nation's spirit by introducing a comprehensive fuel rationing scheme.*

J. E. PAJZS.

UTILIZATION OF HEAT IN INDUSTRIAL FURNACE PLANTS

By HELLMUTH SCHWIEDESEN. (From *Stahl und Eisen*, Vol. 62, No. 8, February, 1942, pp. 149-155).

The General Task of Heat Engineers.

IT is a generally accepted belief amongst experts in Germany that it is quite feasible to reduce fuel consumption by about 10-15% in the manufacture of iron and steel, and by 30-40% in the various fields of the steel manufacturing industry. Such far going saving in fuel would, of course, necessitate the most careful consideration of all possible factors contributing to fuel wastage, no matter how small and unimportant they might appear, since even these relatively minute savings might run up to many million tons of fuel when the whole manufacturing industry is considered. Fuel saving being in wartime of primary importance, it is then the chief task of heat engineers to scan the plant entrusted to him for possible causes of fuel wastage and to stamp them out with the utmost vigour.

The effort of some heat engineers is limited to the use of his knowledge for estimating the proportion of overall costs falling on fuel costs and in doing so he often neglects the far more important task, namely, to reduce fuel wastage. But it also happens in many cases that the efforts of the heat engineer is centered around the right problem, and then his views and those of the production engineer are not concurrent. In such instances final decision rests usually with the production engineer, with the result that changes desirable from a fuel saving point of view are not put into practice. Thus the liability of failing to solve this important problem rests with either the heat engineer or the production engineer.

The engineering knowledge alone of the heating engineer will not suffice in the fight against the wastage—invisible as it is. For really efficient results he has to have a more intimate understanding of the processes taking place; thus a furnace with widely open doors and cracked brickwork should appear to him as a sieve through which water is flowing away; a badly insulated furnace could be compared in a similar manner to a filter which permits water to drain away; excessive heating of a furnace is like a stoker shovelling only half the coal into the furnace and throwing the other half away. When such is the understanding of the heat engineer of the processes connected with fuel wastage then it will be an easy matter for him to carry on with his main duty: to save fuel.

Recent years have borne out a number of publications dealing with fuel saving. It seems, however, pertinent to discuss this problem once more, from yet another point of view, namely by comparing the various types of furnace plants, their efficiency, and the reason for the differential behaviour.

The Proportion of the Total Heating Value of the Fuel turned into Useful Heat in various Types of Furnaces.

The ratio useful heat/total heating value of fuel $\eta_{u/f}$ is a measure of the efficiency of the furnace. If we accept that a fuel saving of 10-15% and more is possible in the steel industry then the fact remains that the proportion of useful heat must be very small indeed, that is, the losses to the walls, cooling media, exhaust gases, etc., are very large.

In fact, a careful analysis has shown that for the manufacture of iron and steel in Germany the overall efficiency $\eta_{u/f}$ is about 50-60% and that for other steel processing only about 20-30%. One may ask, therefore, whether it is permissible at all to anticipate from actual installations an efficiency approximating 1.0?

In practice an efficiency of 1.0 is obtained with a calorimeter. But here the only object is actually to get an efficiency of 1.0, i.e., to determine the total heating value of a particular substance. Turning now to more practical installations, there are a few isolated cases where the efficiency is approximating 1.0. Such is the case, for instance, with autogysers which are used extensively for the production of hot water in households. Their highly efficient performance is due to the fact that in construction they are very similar to calorimeters.

The next nearest approximation to $\eta_{u/f} = 1.0$ is presented in the steel industry by blast furnaces. Although they are often looked upon as high consumers of fuel, in fact they work very economically, if the heating value of the blast furnace gas (which can be utilized) is taken into account, and their efficiency is about 90%.

High efficiency figures can be obtained also with hot blast stoves (80-82%), modern pusher type furnaces, the latter having an efficiency of about 70-73% if fitted with mixture and pressure regulators and economisers. The majority of the pusher type furnaces in operation are obsolete, however, and are working on an average efficiency of about 28-36%.

Still lower efficiency is inherent to Siemens-Martin regenerative furnaces, being of the order of about 28-34% and sometimes even only 20-22%. Exceptionally low is the efficiency of furnaces working at high temperatures without heat regenerators. Such is the case with reheating furnaces, annealing and hardening furnaces. Here, values as low as 15-20% and 8-10% respectively are quite frequently met.

For convenience of analysing the relationship between efficiency and plant design, the efficiency $\eta_{u/t}$ is split up into two components, thus:

(i) The heat received by the furnace/total heat of fuel $\eta_{r/f}$, and

(ii) The useful heat/heat received by the furnace $\eta_{u/r}$.

Both $\eta_{r/f}$ and $\eta_{u/r}$ are separately measures of efficiency, in just the same fashion as their product $\eta_{r/f} \times \eta_{u/r} = \eta_{u/f}$. Of these two partial efficiencies, $\eta_{r/f}$ throws light upon the heating quality of fuel, the efficiency of combustion, the amount of excess of air or fuel as the case might be, the secondary draught and the quality and size of surface exposed to heating. With an electric furnace $\eta_{r/f}$ becomes 1.0 as all the power from the mains is given up as heat in the furnace.

The value of $\eta_{u/r}$ on the other hand, shows the efficiency of the heating surface absorbing useful heat compared to the overall heating surfaces, including the furnace walls, etc.

Improvement of the Ratio, Heat Received by Furnace/Total Heating Value of Fuel.

The ratio $\eta_{r/f}$ can be also expressed as the difference between the total heat of fuel and heat carried away by the exhaust gases, thus:

$$\eta_{r/t} = \frac{\text{Total heat of fuel—sensible heat of exhaust gas}}{\text{total heat of fuel.}}$$

Now in a calorimeter, the heat given up to the water is 100% since the exhaust gas leaves at room temperature. This then represents the ideal case. Up-to-date steam generators and blast heating plants, too, are efficient because the exhaust gases are leaving at about 120 °C and the numerical value of $\eta_{r/f}$ becomes about 93-95%. Pusher type furnaces if properly operated,

have an efficiency $\eta_{r/f}$=83-85%, but under unfavourable conditions, e.g., no air pre-heating, this might fall to 58-60%. Siemens-Martin furnaces operate at about $\eta_{r/f}$ = 55-65%. Reheating furnaces, even under favourable conditions, do not attain higher values than $\eta_{r/f}$ = 35-40%, and an improvement is possible only by using waste heat boilers. Strikingly low efficiency figures are inherent to annealing and hardening furnaces, that is to furnaces which operate at medium temperatures, and although the exhaust gas temperature is moderate, the value of $\eta_{r/f}$ seldom exceeds 50-60%. The main reason for this poor performance is probably that they are run with a high excess of air or gas.

The all important condition for high efficiency $\eta_{r/f}$ is to arrange for the exhaust gas to leave at a low temperature by making it sweep large and efficient heating surfaces which absorb useful heat. This is well illustrated by the efficient operation of steam boilers, blast heating plants and blast furnaces. The first mentioned has a very large heating surface area provided by the drums and tubes. The blast heating plant and the blast furnace, too, have large heating surfaces. A reduction in fuel consumption is possible in the case of the blast furnace also by suitable control of the mixture, which in effect increases the surface swept by the hot gases. This also reduces the carbon monoxide in the exhaust gas together with a reduction of temperature. Further improvement is possible by using waste heat boilers, economisers, and recuperators. Naturally the exhaust gas temperature could also be cooled by cooling elements or by reducing the thickness of the brickwork, but this then would not constitute a useful heating surface, the heat absorbed in it would be wasted.

It would be erroneous to assume that the temperature of the exhaust gas is fixed by the load and working temperature of the furnace ; this holds good only for the main plant, but there is nothing to prevent the engineer from further utilization of the heat content of the ensuing exhaust gases by economisers, recuperators, etc. Unfortunately, however, there are certain limitations to this. With a metallic recuperator for instance, the preheating temperature cannot be higher than 750-800 °C and with brickwork type about 1350-1400 °C. Quantitatively, too, there is a limitation as the pre-heated air, although at the same temperature as the exhaust gas, does not take up all the heat from the exhaust gas.

Optimum utilization of exhaust gases is also hindered by secondary air currents mixing into it and thus cooling it before it enters the recuperator. Another frequent reason for heat losses is leakage of exhaust gas to the atmosphere so that in many instances only about 50-70% of the exhaust gas passes through the recuperator.

From the aforegoing it would be clear that for a high value of $\eta_{r/f}$ the following conditions must be fulfilled :

(1) Employing large and effective heating surfaces (main furnace, economiser, waste heat boiler, etc.).

(2) Proper sealing of system.

(3) Positive control of combustion (without excess of air or gas).

Improvement of the Ratio, Useful Heat/Heat Received by Furnace.

The useful heat comprises the heat quantity taken up by the medium to be heated in the main furnace, economiser, waste heat boiler, etc., as well as the chemical heat content of the gaseous by-product, e.g., blast furnace gas, which is available for further utilization.

Starting again with the calorimeter, here the useful heat is that heat which is absorbed by the water, and once more we get an efficiency $\eta_{u/r}$ approximately 0.1. The blast furnace, too, has a very high efficiency value of about $\eta_{u/r}$ = 95% as very little heat only is lost through the walls. Siemens-Martin furnaces work with an efficiency of no more than 48-50%. Pusher type furnaces are capable of converting 90% of the total heat into useful heat if properly designed, otherwise the efficiency might be as low as 30%.

The critical conditions for high efficiency $\eta_{u/r}$ can be easily determined, and the following considerations should be noted in regard to furnace construction.

Furnaces operated intermittently, such as, for instance, an annealing oven, should not be built from brickwork, partly because they absorb considerable heat quantities when started and partly because they being fairly good heat conductors, in consequence of which much heat would be lost during standby periods. For a similar reason it is not sound practice to insulate a reheating furnace by an auxiliary layer of brickwork. Principally, the following conditions must be fulfilled for high efficiency $\eta_{u/r}$:

(i) Adequate and suitable heat insulation of the furnace plant, taking into consideration the type of operation (whether continuous or not).

(ii) The proportion of useful heating surfaces should be high in proportion to the total heating surfaces. This, in other words, means high load/unit volume (or surface) of furnace.

These conditions are mostly incorporated in steam boilers, blast heating plants, and blast furnaces, hence their high efficiency $\eta_{u/r}$. With pusher type furnaces the ratio of useful heating surface/overall heating surface becomes only 0.2-0.3. Moreover, as heating proceeds, the heat absorbed by the useful heating surfaces decreases, and finally a stage is reached when the heat supplied to the furnace is only required to maintain steady conditions, i.e., to make up for the continuous heat losses, and thus the efficiency at this phase becomes zero. Very similar conditions prevail with annealing furnaces. Consequently, the efficiency $\eta_{u/r}$ of such types of furnaces is markedly lower than that of the previously mentioned types.

Means of Reducing the Useful Heat Demand.

Yet another way of saving fuel is to reduce the heat quantity which is required for a particular job. As an example let us consider the blast furnace.

Here it was found that if the quality of the ore in the mixture charged into the furnace was very poor (i.e., of low iron content), then the coke charge had to be increased from about 900 kg to 1800 kg/t of ore. Remembering that the output of a blast furnace is limited by the coke charge, this means that production is roughly halved. Now if this poor quality ore is prepared before being charged to the furnace by roasting, calcining or sintering then the coke charge could be reduced to about 1300-1350 kg/t ore. Thus considerable saving in coke is possible by reducing the quantity of fuel needed without affecting the efficiency of the furnace. Of course, there is also another side to the question, namely, that all these preparations can be performed only by additional heating, i.e., by further fuel consumption. The important point, however, is that for the latter purpose inferior quality of fuel, such as blast furnace gas can be used, which in some cases would be otherwise wasted, or at any rate is more abundant than coke.

In steel and rolling mill works there is little possibility of saving on the useful heat demand, except by arranging the various phases to follow in quick succession, before the bloom has lost all its heat content from the previous heating operation.

Better possibilities exist for forging plants. Here considerable saving is possible by using high class tools and careful preparation of the work which will cut down the necessary heating operations.

In annealing furnaces some saving is possible by utilizing the heat of the cooling casting for preheating the new one.

Cold rolling, which is being more and more widely used, contributes, too, materially to fuel saving and its wider use should, therefore, be encouraged.

As a summary it can be said that if an overall saving of fuel of 10-40% is to be achieved in the steel industry, as set out in the introduction to this article, then regarding design and operation of furnace plants the following points should be carefully considered:

(1) The application of mixing burners, free from breakdown.

(2) Positive control of air-fuel mixture.

(3) Pressure regulation to prevent hot gases from escaping and secondary air currents from entering the system.

(4) Elimination of all non-essential doors and openings.

(5) Adoption of mechanically operated doors.

(6) Adoption of sluice doors if necessary to cut down heat losses during charging.

(7) More intensive utilization of the heat content of the exhaust gases.

(8) Adequate heat insulation of furnace walls, the coefficient of heat transmission should nowhere exceed 1.0.

(9) Elimination of water cooling elements. Essential ones to be well insulated.

(10) The furnace must be dimensioned to suit average output and not maximum output.

(11) Possible reduction of useful heat demand.

LOW-TIN SOLDERS

By S. TURKUS and A. A. SMITH, Jr., Federated Metals Div., Newark, N.J., and Central Research Laboratory, Barber, N.J. (From *Metals and Alloys*, Vol. 15, No. 3, March, 1942, pp. 412-413).

LEAD-SILVER solders have been known for many years but, because of their relatively high melting temperatures and poor spreading properties, they have not found wide use. It has been found that the addition of tin and bismuth to the lead-silver alloys overcomes these disadvantages to a marked extent and at the same time certain advantageous properties are retained. Due to the urgency of the times, and the importance of conserving available tin, this note is being submitted as a preliminary report in order to bring the promising features of these alloys to immediate attention.

In the accompanying table are given some of the properties that have been determined, using regular tin-lead solders, without silver, for comparison. Carefully standardized testing technique was used for all tests, but space does not permit a detailed description of the methods. It is admitted that considerable additional data are needed before the fields of usefulness of the various alloys can be ascertained.

The ternary alloys of lead-silver-tin offer a variety of solders which have somewhat higher melting points than 40-60 solder but relatively good spreading and bond strength properties. Silver added to low-tin solders lowers the liquidus temperature from 6 to 8 degrees C. and appreciably improves the spreading properties with but minor effect on the bond strength. One outstanding improvement of the silver-bearing solders is the increase in creep resistance as shown in the tabulation. These tests have been in progress for only a few weeks or months, and are far from complete, but the improvement is striking.

Alloys containing 15 Sn, 5 Bi, 1½ Ag, 1% Sb, balance Pb, or 20 Sn, 3 Bi, 1½ Ag, ½ Sb, have liquidus temperatures slightly higher than that of the 40-60 solders, markedly higher tensile strength and a bond strength of approximately 80%. The spreading properties are less than for 40-60 although much greater than those of the binary lead-silver solders. However, it is believed that, except for a few critical uses, this decrease in spreading properties is not of importance in the making of sound joints. By appropriate adjustment of the tin, bismuth and silver contents, fairly wide ranges of melting points and other properties may be obtained. Commercial trials and rapid hand soldering have been made on some of these alloys with excellent results, and automatic machine soldering has also been done with satisfactory results. Cans made on the automatic machine passed all standard tests for 40-60 and 50-50 solders.

Some low-tin solders containing silver, or silver and bismuth, that will surely effect substantial savings in tin consumption have been described in this preliminary note. Both silver and bismuth are available in reasonable quantities and it is probable that ample supplies for solders will be available.

SOME PROPERTIES OF SOFT SOLDERS.

Composition %					Liquidus °C	Solidus °C	Tensile strength lb/sq. in.	Bond strength of lapped joints lb/sq. in.	Spread of ½ Gr. in sq. in.	Creep rate in % per year at 30°C.			
Pb	Sn	Ag	Bi	Sb						200	400	600	800
										lbs/sq. in.			
60	40	—	—	—	238	183	5,660	6,270	1.30	10.9	92.0	300	—
97.5	—	2.5	—	—	304	304	4,980	3,740	0.19	0.10	0.15	—	0.50
95	—	5.0	—	—	375	304	4,915	4,340	0.20	0.18	0.22	—	0.45
90	10	—	—	—	298	183	4,850	4,960	0.27	2.5	12.0	—	—
87.75	10	2.25	—	—	290*	—	4,950	5,000	0.41	—	2.3	8.0	—
80	20	—	—	—	275	183	4,940	5,680	0.37	6.7	18.7	—	—
78	20	2.0	—	—	267*	—	5,620	5,550	0.57	—	—	—	—
70	30	—	—	—	257	183	5,390	5,770	0.83	10.0	50.0	—	—
69	30	1.0	—	—	251*	—	8,810	5,620	0.86	—	4.0	20.0	—
78.5	15	1.5	5	—	264*	—	4,960	5,310	0.47	—	—	—	—
77.5	15	1.5	5	1.0	258*	—	8,000	5,090	0.29	—	—	—	—
74.85	20	1.5	3	0.5	258*	—	8,120	5,380	0.39	—	—	—	—

Note: Bond and spread tests were made on copper sheet.
Tensile and creep tests were made on chill cast strips of the alloys.
*Determined in this investigation from cooling curves—other temperatures from literature.

WIND TUNNELS . . . . BIRTHPLACE OF STREAMLINING

Prepared from Information supplied by DR. F. L. WATTENDORF, U.S. Army Air Corps, Wright Field, and S. PAUL JOHNSTON, Co-ordinator of Research, National Advisory Committee for Aeronautics. (From *Westinghouse Engineer*, Vol. 1, No. 3, November, 1941, pp. 67-70).

ALREADY the Wright brothers used a wind tunnel which contained all the basic ideas of today's tunnels, the principal one being that, aerodynamically, holding a model stationary and blowing air past is equivalent to moving the model through still air. The Wright brothers used a box 22 inches square by five feet long, placed one model after another inside, and fanned air past them at 27 m.p.h. Facts learned with this crude tunnel enabled them to build the plane that left the ground, at Kitty Hawk, in 1903.

Many other engineering works have benefited from wind-tunnel investigations, especially those for which the word "streamline" has any true significance, as high-speed diesel-electric trains and automobiles. Giant skyscrapers, transmission line towers and bridges have been tested as models to prove stability or strength. But it is the airplane that is of surpassing interest at the moment.

The Problem of the Wind Tunnel.

It would be highly desirable to have a wind tunnel capable of testing a full-size plane complete with engine driven propellers, with air moving at maximum flight velocities, as then there would be no need for extrapolating results of the tests. This ideal is ruled out by practical considerations. There exist only two wind tunnels large enough to accommodate an actual-size pursuit plane. One is at Chalais-Mendon, France. The other is at Langley Field, Virginia. The latter belongs to the wind tunnels with an open end. It has a test section of 30 × 60 feet. Air is driven through it by an 8000 H.P. plant at 118 m.p.h. (Fig. 1). To increase the velocity four times—pursuit planes have passed the 400 m.p.h. mark—would require $4^3 = 64$ times the present power input, or 512,000 H.P. Whereas twenty years ago the planes were so slow and small, that the results of wind tunnel tests could be used with only a few corrections, results of tests with models of actual planes could not be used without regard to the results of a thorough investigation of the wind tunnel. It has been found that model test data cannot be considered as strictly correct, if the so-called Reynolds numbers for the test condition and the flight condition are not the same. The Reynolds number is

$$RN = \frac{\text{Air density} \times \text{Air speed} \times \text{Length of object}}{\text{Viscosity of air}}$$

Thus, if the length of the model is less than its flying counterpart, i.e., its "scale" is less, to get equivalent Reynolds numbers the air density or air speed of the test condition must be increased. In practice identical Reynolds numbers can seldom be achieved and correction factors must be applied to test data to scale them up to full size. For many aerodynamic problems identical Reynolds numbers are relatively unimportant.

The second factor affecting the results of model testing comes from the compressibility of air. Any solid object sliding through the air sends ahead of it a pressure wave, which prepares the air for the coming of the object, so that the air can divide and flow freely around it. In normal air this pressure wave travels at about 1120 feet per second (765 m.ph.), that is the speed of sound. If the object runs ahead of its "warning" pressure wave, the effect is shock with loss of energy. The extent of compressibility effect is expressed as the Mach number

$$M = \frac{\text{Air speed}}{\text{Speed of sound}}.$$

Though the speed of present-day aircraft is still only about 400 m.p.h., air may be swirling about some parts of the plane at higher than critical velocity. Actually the Mach compressibility effect makes itself felt at flight speeds above 200 m.p.h., accompanied inevitably by a loss of lift and an increase in power consumption. These problems can be studied in the wind tunnels. Propellers capable of delivering 2000 H.P. to the air have already reached sizes and speeds, in which the tip velocity exceeds sound velocity. These and some other plane appurtenances should, for most usable results, be tested full size.

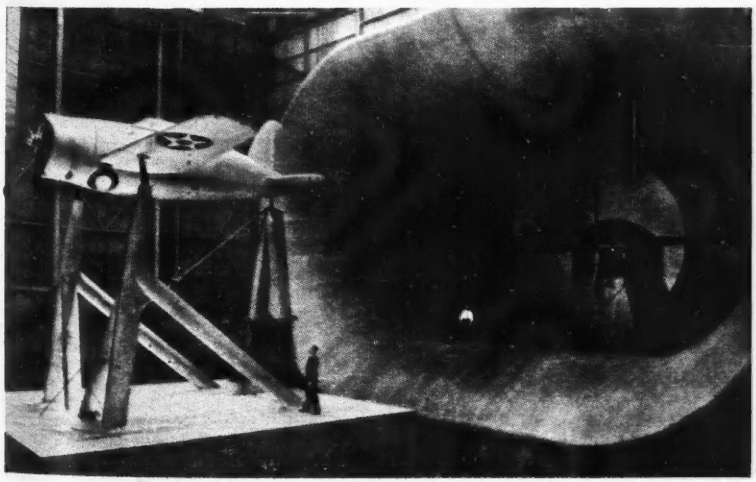


Fig. 1

An actual airplane ready for test in the N.A.C.A. full-scale tunnel at Langley Field. The air stream is 60 feet wide by 30 feet deep. Two 35½ foot propellers, driven by 4,000 H.P. motors produce a wind of 118 m.p.h.

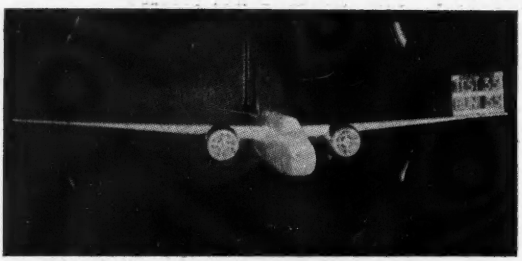


Fig. 2. An eight-foot model of a twin-engine bomber being studied for compressibility effects in the N.A.C.A. 500 m.p.h. wind tunnel.

Several Important Wind Tunnel Types.

Because the scale effect and compressibility effect are unrelated, and because of the new problems introduced especially by military craft with their tremendous engines, gun turrets, bomb racks, and the need of their utmost manœuvreability, there has come a need for a large variety of wind tunnels extant. Several distinct tunnel types have appeared. One is the full-scale wind tunnel, previously mentioned, at Langley Field, of large throat dimensions, but operated at low velocity (118 m.p.h.). High Reynolds numbers can be obtained, actual planes investigated. But it is unsuitable for studies of compressibility effects, because of its low air speed.

Another type is the variable-density tunnel, which is suitable to obtain high Reynolds numbers. A variable density tunnel has been built at Langley Field in 1923, which can be pumped up to 20 atmospheres. Other compressed air tunnels exist in England, Germany and U.S.A. Wright Brothers Tunnel at Massachusetts Institute of Technology, Boston, can be operated at pressures up to 4 atmospheres and under partial vaccum to obtain the equivalent of high altitudes.

Military planes with very high speeds require high-speed wind tunnels. At Langley Field a wind tunnel was built in 1936 with an eight-feet test-section and 500 m.p.h. wind velocity (Fig. 2).

The most powerful high-speed tunnel in the world is the 400 m.p.h. 20 foot tunnel nearing completion (in November, 1941) at Wright Field, near Dayton, Ohio. (Fig. 3). 400 m.p.h. is more than five times the velocity of a 76 m.p.h. wind, called a hurricane. This tunnel is

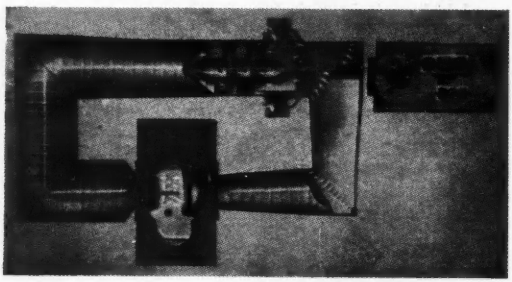


Fig. 3. Wright Field Tunnel for 400 m.p.h. wind velocity. The air is circulated in a counter-lockwise path through the test section (a), by the propellers (b), which are turned by the motor (c) through a long shaft. The motor-generator sets for power recovery and speed control are at (d).

driven by a 40,000 H.P. 300 r.p.m. induction motor with a Kramer cascade. The motor drives two fans in tandem, each with sixteen wooden blades (weighing 1500 lbs. each), through a solid steel shaft 16 inches in diameter and 120 feet long. Dissipation of the heat equivalent to the 40,000 H.P. delivered to the air in the tunnel is achieved partly by natural radiation and convection from the steel housing of the tunnel, partly by exhausting one third of the air each revolution and sucking in an equivalent amount of fresh air. In the Wright Field tunnel there can be tested: 16-foot wing spread airplane models and full scale airplane motors with rotating propellers housed in their normal nacelles. The control room of this tunnel is shown in Fig. 4. The project engineer seated in a sealed air-conditioned control-room can change the attitude of the model at will, without stopping the fan-driving motor. When he presses a button, all instrument readings are simultaneously recorded on tape. The performance of

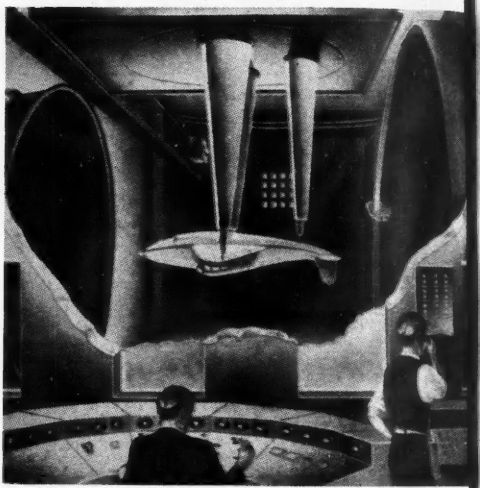


Fig. 4. Control room beside the test chamber of the Wright Field wind tunnel.

any model requires measurement of the six reactions of a body in moving air. Three are forces tending to cause movement in 3 directions and are called lift, drag and side forces. The remaining three are moments tending to turn the body in each dimension. These are pitching (turning end over end), yawning (twisting in a horizontal plane) and rolling (corkscrew) moment. By properly linking the three supports of the model to scales, these three forces and three moments are independently measured in pounds or ounces.

Another even larger tunnel will be built in the near future at N.A.C.A. Ames' Laboratory in California.

Langley Field has 16 wind tunnels of different sizes and air speeds. Most belong to the three classes: full scale, high speed or variable density tunnels or to some combination of them. A 19-foot variable density tunnel is shown in Fig. 5. Another tunnel with a 12-foot test section can be tilted at various angles. It possesses various special means for the control of the plane model. One tunnel is used for the study of tail spins. It is a vertical tunnel with upward flowing air, in which small models are deliberately set in a spin and are photographed. By these means the suitability of various anti-spinning devices can be checked.

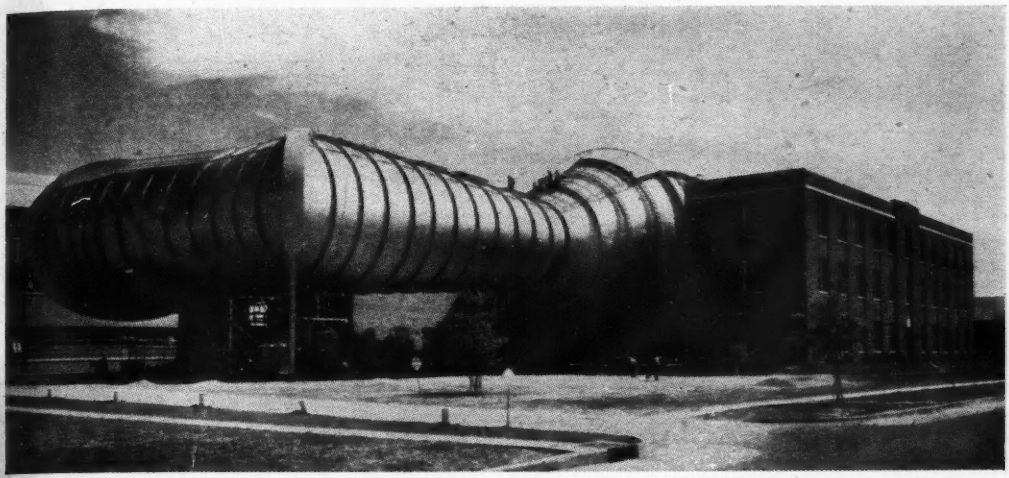


Fig. 5. A 19-foot variable density tunnel at Langley Field.

POWER FOR MAN-MADE HURRICANES

By L. A. KILGORE and J. C. FINK, Westinghouse Electric and Manufacturing Co. (From *Westinghouse Engineer*, Vol. 1, No. 3, November, 1941, p. 71-73.

PROPELLER-DRIVE motors must be capable of wide speed adjustment, usually six or eight to one, or the pitch of the propeller must be adjustable. Variable-speed drives must be able to hold any desired speed nearly constant regardless of supply voltage or frequency fluctuations, at least, for the several minutes required to take a full set of readings. The required accuracy of speed control is usually of the order of 1/4 to 1/2%. The nature of the speed-torque or speed-power curves of a propeller in a tunnel is shown in Fig. 1.

The Direct-Current, Variable-Voltage System (Fig. 2) is one of the oldest and most thoroughly tried methods to obtain speed variation over a wide range. The A.C. driving motor of the motor-generator set is usually a synchronous motor. The field of the generator is varied, thus varying the generator terminal voltage. The motor field requires no adjustment unless variation of the generator voltage is insufficient to give the designed speed range. The variable voltage drive has good speed regulation, with little changes in speed as a result of load fluctuations within the tunnel. The present trend in tunnel designs and the need of larger motors make it increasingly more difficult to apply this driving system, which is restricted to ratings of approximately 4500 H.P. at 650 r.p.m. However, at lower speeds and by using motors with two armatures, the output can be increased.

Wound-Rotor Motor with Liquid Rheostat.

This drive, shown in Fig. 3, comprising a slip-ring motor and an adjustable secondary resistance is very suitable for wind tunnels. The total losses are shown in Fig. 4. They reach a maximum at 2/3 synchronous speed. Great improvements have been made recently in the design of liquid rheostats. The electrolyte is circulated through the rheostat and the heat exchanger; by this means it is kept at 65° C. and no appreciable evaporation occurs. Resistance is changed by variable separation of the electrodes in an electrolyte column of constant area. The current never reaches a density harmful to the electrodes as in the design used previously changing the depth of the immersion of the electrodes. A separate tank is provided for each phase, and the high voltage electrodes are bolted directly to the bronze plates that form the bottoms of the tanks. The tanks are mounted on porcelain insulators, the frame and the moving electrodes are earthed.

One disadvantage of the wound-rotor system is the variation of its torque and speed with the supply voltage. At the lower speeds the resulting per cent. change in speed is nearly equal to the per cent. change in voltage. This limitation can be overcome by connecting to the induction motor shaft a small D.C. machine. By regulating its current with the voltage fluctuations, it provides the necessary additive or subtractive torque to hold speed constant. This combination is shown in Fig. 5. The D.C. machine may be used alone for drives below about 30 per cent. of full speed, and as speed regulating device for higher speeds.

Modified Kramer Set.

Many early variable-speed drives were combined with Kramer sets. They consisted of a synchronous converter connected to the secondary of the wound-rotor motor, feeding power into a D.C. motor on the shaft of the main motor. A modification of this idea, Fig. 6, includes a synchronous motor-D.C. generator set connected to the secondary of the induction motor. The D.C. current power is reconverted to alternating current power at line frequency. This is a very suitable manner to attain economy with large induction motors, when they have to operate at low speed. The Kramer scheme has the advantage that the large induction motor can be started from the D.C. generator connected with it feeding this generator from a small constant-speed set. This scheme has the same inherently good speed regulation as the D.C. variable-voltage system.

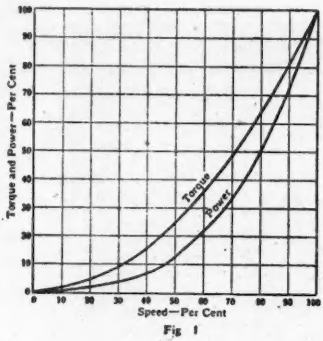


Fig. 1

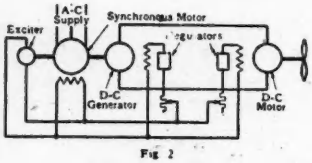


Fig. 2

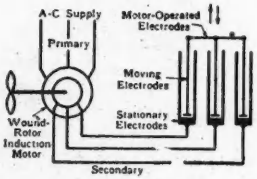


Fig. 3

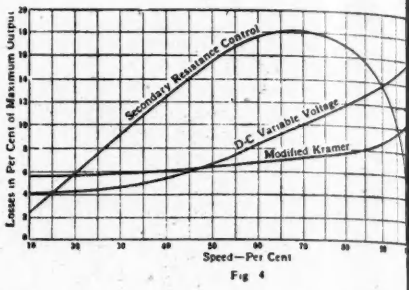


Fig. 4

Fig. 1. Typical power and torque curves of wind tunnel fans; Fig. 2. The basic d.c. voltage drive; Fig. 3. The wound-rotor motor with liquid rheostat for secondary control; Fig. 4. Comparison of losses of different drives of about 5,000 h.p.

Automatic speed control can be readily applied. The modified Kramer scheme is suitable only for the larger installations of say, at least 2000 H.P. induction motors. This is because of the tendency to hunt if the speed is carried too close to synchronism. In a large set with 8 or 10% slip, this hunting tendency can be avoided, but becomes more difficult to handle in units of lesser capacity.

Variable Frequency.

A squirrel-cage induction motor or a synchronous motor could, of course, be supplied with variable frequency over a wide range of speed. The expense of obtaining the variable frequency has limited its use to applications, where the motor speed and capacity are beyond practical limits of D.C. or wound-rotor motors. One method used for supplying variable frequency consists (Fig. 7) of a combination of an A.C. to D.C. and a D.C. to A.C. motor generator set. Some installations have provision for changing the number of poles on the variable speed synchronous generator and on the propeller driving synchronous motor.

Another means considered, shown in Fig. 8, consists of a large wound-rotor induction machine used as a frequency changer. The high slip frequency in the rotor at maximum current creates difficult design problems when large capacities are required. Also, it necessitates a variable speed drive for the frequency changer. This is a large D.C. motor and a D.C. to A.C. set of slightly greater capacity for the full range of speed than for the modified Kramer set because the total losses of the propeller-drive motor must be converted. The speed of the variable-speed set can be

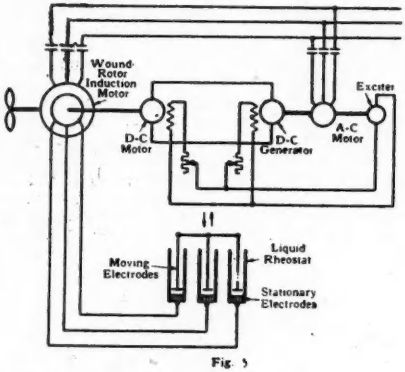


Fig. 5

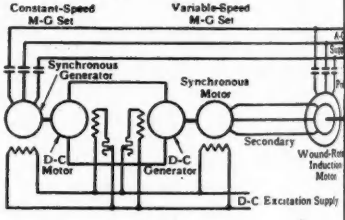


Fig. 6

Fig. 5. A wound-rotor motor and liquid-rheostat control with a d.c. speed regulator to obtain a speed below that obtained by control of the motor secondary resistance.

Fig. 6. The modified Kramer drive, with a wound-rotor driving motor and two motor-generator sets to return some of the secondary losses to the line. Speed is controlled over a wide range by control of the fields of both d.c. machines.

Fig. 7. Variable frequency can be supplied through two motor generator sets to a main synchronous motor.

Fig. 8. Variable frequency can also be supplied by a wound-rotor induction frequency changer system.

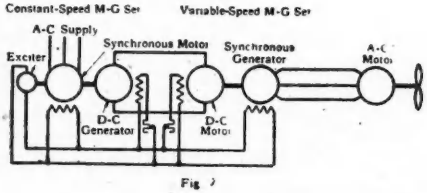


Fig. 7

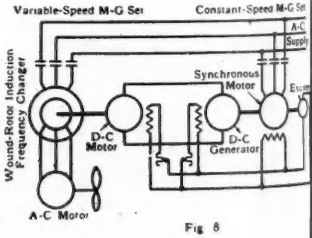


Fig. 8

kept high and the size reduced by taking advantage of the fact that the high current occurs only at low speed, just as for the modified Kramer set.

Multi-Speed Motor with Variable-Pitch Propeller.

A 2000 H.P. drive was built in 1938 with a four speed squirrel-cage motor and a variable pitch propeller (Fig. 9). Using two complete windings in the slots, each with two connections for a two-to-one change in speed, it is possible to obtain speeds corresponding to 6, 8, 12 and 16 poles. The pitch of the propeller blades is varied under load to give intermediate air speeds. A high efficiency is obtained through most of the range. This particular tunnel has a range in pressure from 1/4 to 4 atmospheres and a corresponding range of 16 to one in density, producing a wide range of load at the various speeds. The chief limitations of this scheme are the difficulty and expense in variable-pitch mechanism for larger ratings and the relatively high starting currents required.

The final choice of the type of drive is a balance between many factors. For large installations where extensive use justifies the investment on an efficiency basis, the modified Kramer set probably remains the choice. For smaller units at high efficiency, either the D.C. variable-voltage or the multi-speed motor with variable-pitch propellers seems the better choice. Often, however, where the probable use will not justify the investment, the wound-rotor motor and liquid rheostat secondary control should be used. If the range or accuracy is beyond that attainable with simple rheostat control, the addition of a small auxiliary D.C. machine with suitable control will be the answer.

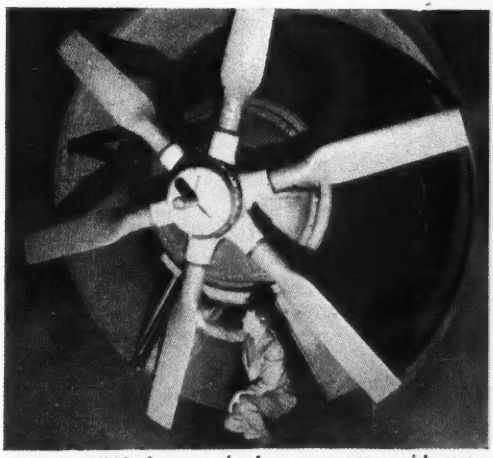


Fig. 9. 2000 h.p. squirrel cage motor with two windings driving variable pitch propeller.

PORT WASHINGTON ESTABLISHES FIVE-YEAR ECONOMY RECORD

(From *Electrical World*, Vol. 116, No. 6, August 9th, 1941, pp. 70-73).

PERFORMANCE records of Port Washington show that this plant has held for five years, since it was started at the end of 1935, the title of "Worlds' most efficient Station."*

Port Washington is an 80,000 kW single-unit plant owned by the Wisconsin Electric Power Company, and is located on Lake Michigan, 28 miles North of Milwaukee. Salient features of the plant which operates on the reheat cycle are:

One 690,000 lb. per hour combustion engineering boiler delivering steam of 1,230 psi. (86.5 at.) and 825 F. (440° C.). Powdered coal having 13,000 B.t.u. per lb. is burned.

One 80,000 kW Allis Chalmers tandem-compound turbine, 1800 r.p.m. with air-cooled generator, 22,000 volt.

Work is currently in progress on a second 80,000 kW turbo-generator with hydrogen-cooled generator, and another 690,000 lb. per hour boiler. An improvement of the efficiency of the plant by this second unit is expected.

*This title may go over for the next time to the Twin Branch Station of the American Gas and Electric Company, located near South Bend, Ind. (Fully described in 8 articles in *Electrical World*, Vol. 116, No. 16, Oct. 18th, 1941, p. 87/106). This power station has been started in May, 1941, with one unit of 75,500 kW, steam pressure **2500 psi** (176 at.) 950 F (510° C).

Thermal Performance	May 12th to June 6th	June 24th to 30th.
Average kW generated per hour	51,757	66,024
B.t.u. per kW-hour output ..	10,282	**10,189**
Thermal Efficiency % ..	33.18	**33.49**

FIVE YEAR PERFORMANCE RECORD

Station Heat Consumption B.t.u./kW.-hour	Annual Averages 1940	1939	1938	1937	1936	5-Year Average
Gross	10,182	10,198	10,217	10,277	10,376	10,228
Auxiliaries	547	572	571	558	578	565
Net	**10,729**	10,770	10,788	10,835	10,954	10,813
Load Factor %	65.8	59.0	59.3	58.4	57.5	60.0
Availability factor %	**89.4**	88.7	95.0	86.4	89.8	89.9
Plant operated hours % of total ..	89.4	88.7	91.0	84.4	86.7	**88.04**
Mechanical trouble hours % of total ..	9	—	—	1.5	3.4	1.15
Corrosion trouble hours % of total ..	0.9	2.5	—	4.5	1.5	1.88

THE INFLUENCE OF COMPRESSIBILITY OF THE FLUID ON THE PROPERTIES OF A CENTRIFUGAL BLOWER.

By A. MELDAHL. (From *The Brown Boveri Review*, Vol. 28, Nos. 8-9, Aug./Sept., 1941, pp. 200-202).

IN modern high-speed blowers the air is already considerably compressed in the wheel itself. The characteristic of the wheel is thereby changed to an appreciable extent and blowers differ in this point from low-speed ventilators in which, as in pumps, the fluid has practically always the same volume.

According to the laws of similarity, the compressibility is expressed by the number of Mach, i.e., by the ratio of any velocity to the velocity of sound in the fluid. With blowers, it is convenient to compare the peripheral velocity u_2 to the velocity of sound a_o in the fluid at rest on the suction side of the blower. We consequently define:

$$M_o \equiv \frac{u_2}{a_o} \qquad .. \qquad .. \quad (1)$$

In principle, the influence of the number of Mach can only be found by experiment. With certain assumptions, however, some of the consequences of compressibility may also be calculated.

(*a*) *Influence of Compressibility on the Power Consumption of a Blower Wheel.*

The theoretical manometric head of the blower is $H_o = L/G$ (L = power, G = quantity), from which a part $H_2 = \rho_o \cdot H_o$ is transformed into pressure in the wheel itself, with the efficiency $\eta_R \cdot \rho_o$ is the theoretical degree of reaction and may be calculated from observations. From this, the temperature T_2 at the outer circumference of the wheel is found:

$$\frac{T_2}{T_o} = 1 + \frac{k-1}{k} \frac{\rho_o H_o}{R T_o} \qquad .. \qquad .. \quad (2)$$

and with polytropic compression the volume in the wheel diminishes in the proportion

$$\frac{v_o}{v_s} = \left[1 + \frac{k-1}{k} \cdot \frac{\rho_o \cdot H_o}{R T_o} \right]^{\frac{1}{n-1}} \qquad .. \quad (3)$$

n is the polytropic coefficient of compression in the wheel, which can also be replaced by the adiabatic coefficient k and the wheel efficiency η_R, according to the equation $(k-1)/k = \eta_R (n-1)/n$. If, instead of the theoretical manometric head H_o, a dimensionless power coefficient μ_o be introduced according to the definition $H_o = \mu_o u_2^2/g$ and if it be further considered that $u_2^2 / kg\, R\, T_o = u_2^2 / a_o^2 = M_o^2$, the following equation is found:

$$\frac{v_o}{v_s} = \left[1 + \frac{k-1}{k} \cdot \frac{\rho_o H_o}{R T_o} \right]^{\frac{k}{k-1} \cdot \eta_R - 1} =$$

$$= \left[1 + (k-1)\, \rho_o\, \mu_o\, M_o^2 \right]^{\frac{k}{k-1} \eta_R - 1} \quad .. \quad (4)$$

The equation shows that the compression ratio is, in fact, only a function of the number of Mach, respectively of the equivalent expression H_o/RT. As it was assumed that conditions at the circumference and consequently the power consumption, respectively the theoretical manometric head H_o, will not change as long as the volume V_s at the outer circumference of the wheel remains constant, the suction volume v_o must be increased in the ratio v_o/v_s. If, therefore, the curve $H_o = f(V_o)$ is known for an incompressible fluid, the corresponding curve for the compressible fluid will be found by deplacing every point at the same value of H_o in the proportion v_o/v_s in the direction of the V_o axis.

Fig. 1 shows the influence of compressibility. The power consumption increases considerably with the number of Mach. At the limit volume the curves break down vertically, the dotted parts cannot be realized in practice.

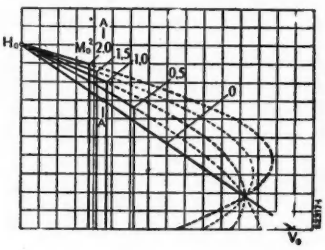


Fig. 1
Theoretical manometric height of a wheel with different numbers of Mach.
H_o. Theoretical manometric height.
V_o. Suction volume
A—A. Normal volume.

Based on the straight line for $M_o^2 = O$, the lines for $M_o^2 = 0.5$, 1.0, 1.5 and 2.0 were calculated.

At the limit volume, which may be taken from Fig. 2 these lines break down vertically. The real manometric height is found by multiplying the theoretical manometric height by the efficiency.

(*b*) *Limit Volume.*

With high-speed compressors for heavy gases it sometimes occurs that the pressure volume curve breaks down almost vertically, a little above the normal point, because the speed of sound is attained in the wheel entrance and, therefore, no bigger volume can enter the wheel. This limiting volume may be calculated in the following way:

Let w_1 be the relative entrance velocity and u_1 the corresponding peripheral speed. Then the increase of energy in the entrance of the wheel is :

$$H_1 = (u_1^2 - w_1^2)/2g \quad .. \quad (5)$$

As soon as $w_1 > u_1$ expansion takes place in the entrance, which increases the specific volume and diminishes the temperature and consequently the velocity of sound.

The following equations then hold :

$$\frac{T_1}{T_o} = 1 + \frac{k-1}{k} \cdot \frac{H_1}{R\,T_o} = 1 + (k-1)\frac{(u_1^2 - w_1^2)}{2a_0^2} \quad .. \quad (6)$$

and

$$\frac{v_o}{v_1} = \left[1 + (k-1)\frac{(u_1^2 - w_1^2)}{2a_0^2}\right]^{\frac{1}{n-1}} \quad .. \quad (7)$$

If the speed of sound is attained in the entrance, then $w_1^2 = a_o^2 \cdot T_1 / T_o$, and consequently :

$$\frac{T_1}{T_o} = \frac{2 + (k-1)\, u_1^2 / a_0^2}{k+1} \quad .. \quad (8)$$

and

$$\frac{v_o}{v_1} = \left[\frac{2 + (k-1)\, u_1^2 / a_o^2}{k+1}\right]^{\frac{1}{n-1}} \quad .. \quad (9)$$

The maximum quantity which may enter the wheel per unit section of free entrance is a_1/v_1 , whereas the corresponding quantity with incompressible fluid would be $u_1 / v_o \cos \alpha_1$, with shockless entrance under the angle α_1 .

The ratio of limit volume to the shockless normal volume with incompressible fluid consequently is :

$$\frac{V_{max}}{V_o^*} = \frac{v_o}{v_1} \cdot \frac{a_1}{\mu_1} \cdot \cos \alpha_1 \quad .. \quad (10)$$

or with $a_1^2 = a_0^2 \cdot T_1/T_0$ and $u_1 = u_2\, r_1/r_2$

$$\frac{V_{max}}{V_o^*} = \frac{\cos \alpha_1}{M_o\, r_1/r_2}\left[\frac{2 + (k-1)\,(M_o\, r_1/r_2)^2}{k+1}\right]^{\frac{n}{n-1}\frac{1}{2}} \quad (11)$$

If losses must be considered (n-1)/n would have to be replaced by $\eta_1\ (k-1)/k$, as the fluid expands in the entrance when operating with limit volume if u_1 is smaller than a_1 . Calculation shows that the ratio V_{max}/V_o may become less than unity.

The reason is that the quantity which may enter the wheel without shock decreases with increasing number of Mach. Let V_o' be the entrance volume without shock, referred to suction conditions. Then

$$\frac{V_0'}{V_0^*} = \left[1 - \tfrac{1}{2}(k-1)\,\alpha_1\,(M_0\, r_1/r_2)^2\right]^{\frac{1}{n-1}} - 1 \quad (12)$$

This decrease in volume, entering without shock, is comparatively small, and is only of importance in high-speed refrigerating compressors.

In Fig. 2 the ratio of the limit volume to the normal volume entering without shock and with incompressible fluid is shown in function of the number of Mach. When M_o approaches unity, the limit volume has already decreased to the normal volume, when M_o is bigger than unity, the normal volume cannot be attained at all.

The ratio of volumes that may enter the wheel without shock with compressible and incompressible fluid is also shown, for $M_o \sim 1$, the decrease is about 10%.

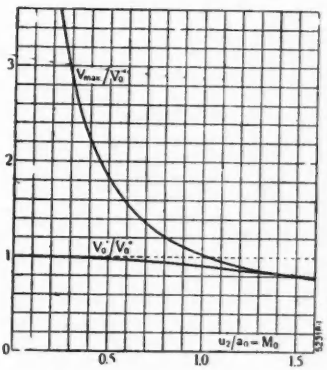


Fig. 2

Variation of the limit volume and the volume entering without shock with the number of Mach.

V_{max}. Limit volume V_o'. Volume entering without shock with compressible fluid.

V_o^*. Volume entering without shock with incompressible fluid (i.e., "Normal volume").

The limit volume decreases considerably with increasing number of Mach and in the present example reaches the "normal volume" at $M_o \sim 1$. It may be increased by a suitable design of the wheel.

(c) Influence of Compressibility on Diffusor Dimensions.

At constant suction volume the diffusor must be closed considerably with increasing compression, i.e., with increasing number of Mach. The volume leaving the wheel diminishes considerably and also in the intermediate space between wheel and diffusor blading compression will steadily increase. For wheels with falling characteristic, the absolute outlet volume and consequently the diffusor must be still more closed.

If H_{2kin} is the kinetic energy at the wheel outlet, then in an unbladed diffusor with parallel walls, the fraction $H_{2kin}\ [1 - (r_2/r_3)^2]$ will be converted into pressure if r_2/r_3 represents the ratio of wheel radius to the entrance radius in the diffusor.

Analogously to equation (4), the compression ratio between wheel and diffusor then becomes :

$$\frac{v_2}{v_3}\left\{1+(k-1)\frac{1-(r_2/r_3)^2}{1+(k-1)\rho_o\mu_o M_o^2}\cdot \frac{(1-\rho_o)\mu_o M_o^2}{\eta_s}\right\}^{\frac{1}{k-1}\eta_s-1} \quad (13)$$

with η_s = diffusor efficiency between wheel circumference and diffusor blading.

The total compression until entrance in the diffusor is then $v_o/v_3 = v_2/v_3 \cdot v_o/v_2$, and with constant tangential component of the absolute outlet velocity the diffusor must be closed in this ratio with increasing number of Mach.

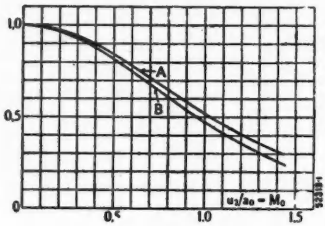


Fig. 3

Variation of diffusor section with the number of Mach.

A. For delivery of normal suction volume.

B. When volume enters wheel without shock.

With increasing number of Mach, the volume entering the diffusor diminishes considerably, and with wheels having blades bent backwards, the angular momentum increases. For both reasons the diffusor sections must be diminished. The figure shows that this decrease may be considerable; in the present example it is about 50% for $M \sim 1$.

Fig. 3 shows the result of calculation in one special case, the theoretical manometric height being taken from Fig. 1. The decrease in diffusor dimensions is considerable, and approaches 50% for $M \sim 1$. A still greater decrease is necessary if it is considered that according to Fig. 2 the volume which may enter the wheel without shock, also diminishes with increasing number of Mach.

In this connection the question may be raised under which conditions the local velocity of sound is attained in the entrance of the diffusor blading. The velocity of sound in the entrance of the diffusor is

$$a_3^2 = a_o^2 + g(k-1)(H_o - H_{3\,kin}) \quad .. \quad (14)$$

and on the other hand the velocity of entrance is

$$c_3^2 = 2g H_{3\,kin} \quad .. \quad .. \quad (15)$$

The velocity of sound is attained if $c_3^2 = a_3^2$, that is if

$$a_o^2 = (k+1)g H_{3\,kin} - (k-1)g H_o \quad .. \quad (16)$$

For H_{3kin} the following equation holds:

$$H_{3\,kin} = \left[1 - \frac{1-(r_2/r_3)^2}{\eta_s}\right](1-\rho_o)\mu_o H_o \quad .. \quad (17)$$

and as with normal dimensions

$$1 - \frac{1-(r_2/r_3)^2}{\eta_s} \sim {}^2/_3$$

the condition that the local velocity of sound is attained in the diffusor entrance, is:

$$M_o^2 \sim \frac{1}{\mu_o[{}^2/_3(k+1)(1-\rho_o)-(k-1)]} \quad (18)$$

In an ordinary blower for air, the velocity of sound is never attained in the diffusor entrance. With wheels with radial blading the limit lies at $M_o \sim 1.6$, that is in the neighbourhood of $u_2 = 500$ m/s. Refrigerating compressors for heavy refrigerants with $k \sim 1$, attain velocity of sound in the diffusor entrance at about the same value of $M_o \sim 1.6$, i.e., at a peripheral velocity of about 200-250 m/s, if the much lower speed of sound in these refrigerants is considered.

THE SEPARATION OF IMPELLER AND DIFFUSOR LOSSES IN RADIAL BLOWERS.

By A. MELDAHL. (From *The Brown Boveri Review*, Vol. 28, Nos. 8-9, Aug./Sept., 1941, pp. 203-206).

IT is not easy to establish a true definition of what the impeller efficiency really is, in the case of a radial blower. The impeller certainly absorbs all the power supplied but only converts a part of it into pressure while the kinetic energy it delivers is converted into pressure in the diffusor.

By means of the following consideration it is possible to reach a workable definition which has also the advantage that the efficiency thus defined is relatively easy to measure.

A blower generates pressure. Now assuming the same power input and the same delivery quantity, the better the blower the higher the pressure it will generate. Thus, it seems best to compare the real blower with an ideal one, without losses, delivering the same quantity and absorbing the same power. According to Euler's formula, the tangential component of the absolute delivery velocity at the periphery of the impeller then is the same in the ideal and in the real blower. If the ideal blower be so dimensioned that the radial component is the same as well, the kinetic energy in the gap will be identical for real and for ideal blower.

It is easy to calculate the pressure generated in the ideal blower. The total theoretical delivery head is $H_o = L/G$ (L = power input; G = weight delivered per unit of time); with inflow without angular momentum, the tangential component of the absolute delivery velocity at the

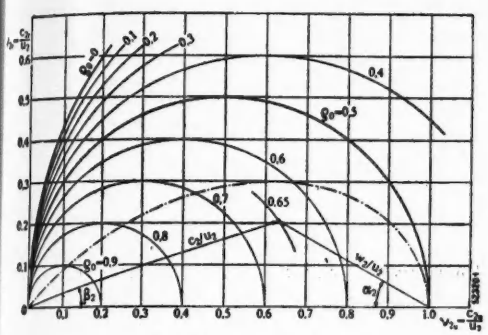


Fig. 1.—v_{2u}—v_{2r}. Diagram with lines of equal degree of reaction in the case of inflow devoid of angular momentum.

The dash-dot-line is that of maximum degree of reaction for a given discharge angle β_2.

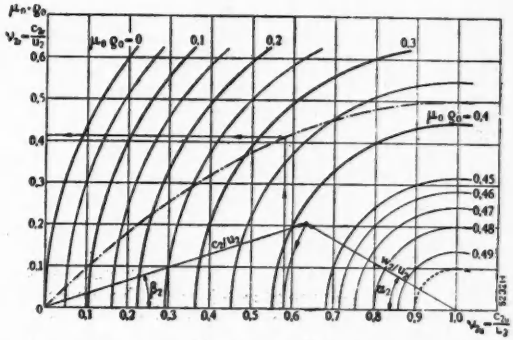


Fig. 2.—v_{2u}—v_{2r}. Diagram with lines of equal theoretical pressure in the gap, with inflow devoid of angular momentum.

The theoretical pressure in the gap $\mu_o \rho_o$ can also be read off the left-hand scale as shown by the arrows.

$v_{2u}=c_{2u}/u_2$. Tangential component of the absolute velocity of discharge.
$v_{2r}=c_{2r}/u_2$. Radial component of the absolute velocity of discharge.
α_2. Absolute angle of discharge. β_2. Relative angle of discharge.

The diagram is a dimensionless representation of the discharge triangle.

impeller periphery, is $c_{2u} = gHo/u_2$ (u_2 = peripheral speed); the radial component is $c_{2r} = G.v_s/\pi D_2 b_2$ (v_s = specific volume in gap, D_2 and b_2 = diameter and width of impeller). Thus, the velocity head in the gap is $H_{2kin} = 1/2\, g\,(c_{2u}^2 + c_{2r}^2)$ and the delivery head in the ideal impeller is $H_2 = H_o - H_{2kin}$.

The real blower generates a delivery head H, and, therefore, the efficiency of the blower is $\eta = H/H_o$. This delivery head can be divided up between the impeller and the diffusor when the static gap pressure on the impeller periphery is measured. Let the actual delivery head up to the gap be H_s. Then the diffusor efficiency is obviously

$$\eta_D = \frac{H - H_s}{H_{2kin}}$$

Analogously the impeller efficiency will be defined as

$$\eta_R = \frac{H_s}{H_2} = \frac{H_s}{H_o - H_{2kin}} \quad .. \quad (1)$$

this definition leads immediately to

$$\eta = \eta_R\, \rho_o + \eta_D\,(1 - \eta_o) \quad .. \quad (2)$$

here $\rho_o = H_2/H_o$ is the theoretical degree of reaction. This simple equation (2) for η shows that the definition according to (1) is a useful one.

With the help of the theoretical degree of reaction, the efficiencies for impeller and diffusor can be expressed as

$$\eta_R = \frac{H_s}{\rho_o H_o}; \quad \eta_D = \frac{H - H_s}{(1-\rho_o) H_o} \quad .. \quad (3)$$

It is advantageous to introduce here dimensionless characteristic numbers, all delivery heads being referred to u_2^2/g. Then $H_o = \mu_o u_2^2/g$, $H = \mu u_2^2/2g$ and $H_s = \mu_s u_2^2/g$ which leads to

$$\eta_R = \frac{\mu_s}{\rho_o \mu_o}; \quad \eta_D = \frac{(\mu - \mu_s)}{(1-\rho_o)\mu_o} \quad .. \quad (4)$$

ρ_o has still to be found. According to the Euler formula $c_{2u} = g\, H_o/u = v_{2u}\,.\,u_2$ for inflow without angular momentum. The Rateau volume figure is introduced for the volume according to the equation $V = \delta\,.\,\frac{1}{4} D_2^2\, u_2$, therefore $c_{2r} = \delta_s\,.\,\frac{\pi}{4}\,.\,D_2/b_2\,.\,u_2 \equiv v_{2r}\,.\,u_2$ (δ_s is referred to the volume in the gap) and this leads to

$$\mu_o \rho_o = \mu_o - \tfrac{1}{2}(v_{2u}^2 + v_{2r}^2) \quad .. \quad (5)$$

or

$$\rho_o = 1 - \frac{(v_{2u}^2 + v_{2r}^2)}{2\mu_o} \quad .. \quad .. \quad (6)$$

If the inflow is without angular momentum, Euler's formula leads to $v_{2u} = \mu_o$ and then both expressions can be graphically expressed very simply in a system of co-ordinates v_{2u}/v_{2r} (Figs. 1 and 2). The lines $\mu_o \rho_o$ = constant are concentric circles round the point ($v_{2u}=1$; $v_{2r}=0$) and with a radius $r = \sqrt{1 - 2\,\mu_o \rho_o}$. The curves ρ_o = constant are also circles which all pass through the zero point ($v_{2u}=0$; $v_{2r}=0$) and the centres of which lie on the v_{2u} axis; the radii are $r = 1 - \rho_o$.

v_{2r} is proportional to the delivery volume in the gap, therefore, the $v_{2u} - v_{2r}$ curve at inflow without angular momentum is nothing else than dimensionless theoretical pressure-volume curve $\mu_o = f(\delta_s)$ somewhat deformed in the direction v_{2r}.

The curves $\mu_o \rho_o$ = constant and ρ_o = constant can be transferred directly to the pressure-volume diagram, the circles becoming ellipses. The $v_{2u}-v_{2r}$ diagram, it may be mentioned in passing, is nothing else than a dimensionless representation of the outflow triangle.

The calculation of the efficiency of the impeller from the pressures measured, power inputs and delivery quantities is then carried out as follows: In order to determine the radial velocity in the gap, the specific volume must be known. The pressure p_s is measured. The temperature is computed as follows: The theoretical delivery head of the impeller up to the gap is $\mu_o \cdot \rho_o \cdot u_2^2/g$; $\rho_o \cdot \mu_o$ and ρ_o respectively is still an unknown quantity, but can be estimated very closely, as soon as the outlet triangle is approximately known. The temperature in the gap is, therefore, given by

$$T_s = T_o\left[1 + \frac{k-1}{k} \cdot \frac{H_o}{R\,T_o}\,\rho_o\right]$$

$$= T_o\left[1 + (k-1)\,\mu_o\,\rho_o\,M_o^2\right] \quad \ldots \quad (7)$$

Here $M_o = u_2/\sqrt{kgRT_o}$ is the Mach number. The specific volume in the gap amounts to $v_s = RT_s/p_s$ and the radial component of the outflow velocity is $c_{2r} = Gv_s/\pi D_2 b_2$, and from this $v_{2r} = c_{2r}/u_2$ may be calculated.

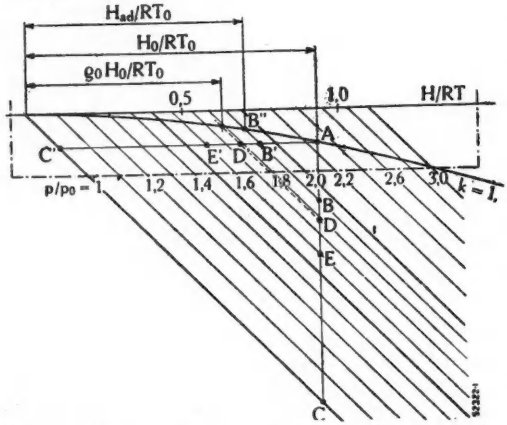


Fig. 3. Diagram to determine the efficiencies from the pressures measured.

A. Theoretical pressure corresponding to the delivery head H_o.
B. Pressure measured. Efficiency of blower: $\eta_{pol} = BC/AC = B'C'/AC'$.
D. Theoretical pressure in gap corresponding to the delivery head H_o.
E. Pressure measured in gap.
Impeller efficiency: $\eta_{R\,pol} = CE/CD = C'E'/C'D'$.
Diffusor efficiency: $\eta_{D\,pol} = BE/AD = B'E'/A'D'$.

The peripheral component of the delivery velocity of the impeller can be calculated from the power input. In the case of inflow without angular momentum $H_o = L/G$ and $v_{2u} = g\,H_o/u_2^2 = \mu_o$.

$\mu_o \rho_o$ can now be determined exactly, it is hardly necessary to repeat the calculation for v_{2r}.

The polytropic work of compression in the impeller is

$$H_s = R\,T_o \cdot \frac{n}{n-1}\left[\left(\frac{p_s}{p_o}\right)^{\frac{n-1}{n}} - 1\right] \quad (8)$$

from this we deduce $\mu_s = g\,H_s/u_2^2$ and from this $\eta_R = \mu_s/\mu_o\,\rho_o$. Unfortunately, the polytropic coefficient n depends on η_R so that n must first be assumed and then eventually corrected. This can be avoided, however, for the ideal comparative compressor we have:

$$H_2 = R\,T_o \frac{k}{k-1}\left[\left(\frac{p_2}{p_o}\right)^{\frac{k-1}{k}} - 1\right] \quad \ldots \quad (9)$$

where the ideal gap pressure $p_2 > p_s$. By dividing (8) by (9) we get

$$\frac{H_s}{H_2} = \eta_R = \frac{n}{n-1} \cdot \frac{k-1}{k} \cdot \frac{\left[\left(\frac{p_s}{p_o}\right)^{\frac{n-1}{n}} - 1\right]}{\left[\left(\frac{p_2}{p_o}\right)^{\frac{k-1}{k}} - 1\right]} \quad (10)$$

As now $$\frac{n}{n-1} \cdot \frac{k-1}{k} = \eta \quad \ldots \quad \ldots \quad (11)$$

therefore $$\frac{p_s}{p_o} = \left(\frac{p_2}{p_o}\right)^{\eta_R} \quad \ldots \quad \ldots \quad (12)$$

or

$$\eta_R = \frac{\log\,(p_s/p_o)}{\log\,(p_2/p_o)} \quad \ldots \quad \ldots \quad (13)$$

The ideal gap pressure ratio p_2/p_o can be deduced from equation (9)

$$\frac{p_2}{p_o} = \left(\frac{T_s}{T_o}\right)^{\frac{k}{k-1}} = \left[1 + \frac{k-1}{k} \cdot \frac{H_o}{R\,T_o} \cdot \rho_o\right]^{\frac{k}{k-1}} =$$

$$= \left[1 + (k-1)\,\mu_o\,\rho_o\,M_o^2\right]^{\frac{k}{k-1}} \quad (14)$$

Analogously for the diffusor we get

$$\eta_D = \frac{\log\,(p/p_s)}{\log\,(p_{th}/p_s)} \quad \ldots \quad \ldots \quad (15)$$

where p_{th} is the theoretical delivery head of the blower, hat is at efficiency = 100%. For the entire blower

$$\eta = \frac{\log\,(p/p_o)}{\log\,(p_{th}/p_o)} \quad \ldots \quad \ldots \quad (16)$$

These expressions are so simple because the gap temperature T_s is the same for the ideal and for the real blower.

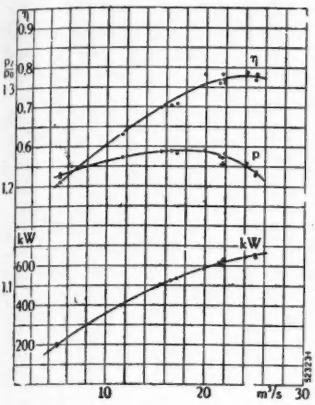
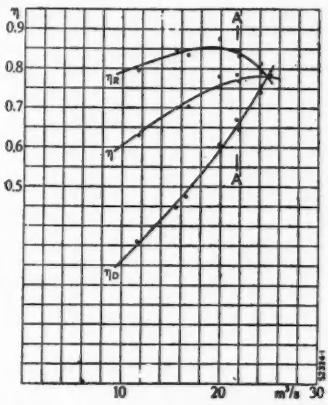


Fig. 4a (left).
Characteristic of a single stage blower. KW = power, P = Pressure, η = Efficiency, + = Normal point.

Fig. 4b (right)
Impeller and diffusor efficiencies. η_R = Impeller efficiency, η_D = Diffusor efficiency, η = Blower efficiency, A—A = Normal point.



These formulæ can serve as basis for a very simply diagram. The expression

$$f(x;k)=\frac{1}{k-1}\log_e\left(1+\frac{k-1}{k}.x\right)-x$$

can be represented graphically, as a function of x with k as parameter. For $k = 1$ (isothermal compression), $f(x;1) = 0$. Further the lines $\log_e(p/p_o)$ -x are drawn in for a suitable series of values p/p_o.

The utilization of the diagram is shown in Fig. 3 for the sake of simplicity with only one curve corresponding to $k = 1.4$. If the theoretical delivery head $H_o = L/G$ is known then we have $AC = \log p_{th}/p_o$ in the diagram. The actual pressure ratio attained is plotted in B then $\eta_{Pol} = BC/AC$. The work of compression in the impeller is $\rho_o H_o$ and, therefore, the length $CD = \log(p_2/p_o)$. The pressure ratio measured in the gap is plotted in E so that $\eta_{Rpol} = CE/CD$. For the diffusor analogously $\eta_{Dpol} = BE/AD$.

By an artifice the diagram can be made more handy. It is at once evident that $AC = AC'$ and $AB = AB'$. It is, therefore, more practical to draw a horizontal line through A to C' and to plot the pressure really measured at B'. Then $\eta_{pol} = B'C'/AC'$; $\eta_{Rpol} = C'E'/C'D'$ and $\eta_{Dpol} = B'E'/AD'$ The whole lower part of the diagram can thus be eliminated.

The illustrations Figs. 4 and 5 demonstrate the practical value of the separate determinations of the efficiencies. Fig. 4a shows the pressure curve, power input curve and efficiency curve of a blower; the character of these curves is quite normal. In Fig. 4b, however, the efficiencies of impeller and diffusor are represented separately. These curves show clearly that the diffusor is much too big, because even with the biggest amount of air delivered the maximum value is not attained.

Therefore, the blower was fitted with a small diffusor, and the success of this modification is shown by Fig. 5a. The efficiency rose by 3% and the character of the p-v curve was improved as well. Fig. 5b shows that the efficiencies of the impeller and the diffusor now attain their

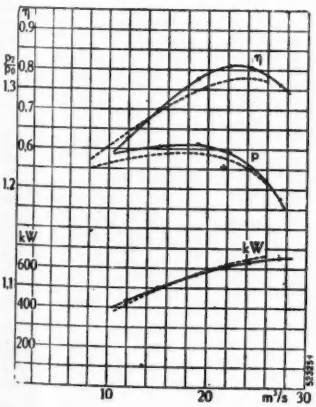
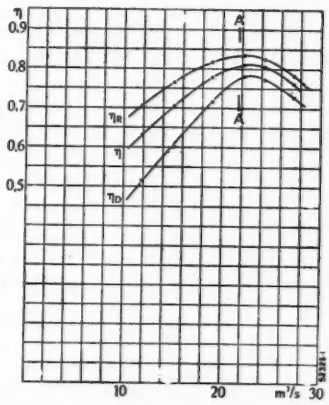


Fig. 5a (left).
Characteristics of the same blower with a small spiral housing.
KW = Power, P = Pressure, η = Efficiency, + = Normal point, - - - - = Curves for original spiral housing.

Fig. 5b (right).
Efficiencies of impeller and diffusor. η_R = Efficiency of impeller, η_D = Efficiency of diffusor. η = Total efficiency, A—A = Normal point.



maximum value simultaneously at the normal point, as, indeed, should be the case.

The separate measurements of the efficiencies of impeller and diffusor is, therefore, a very useful means towards further perfecting blowers and compressors.

THE DETERMINATION OF THE VARIATION OF STATE IN TURBO-MACHINERY BY MEANS OF THE INCREASE IN ENTROPY.

By O. ZWEIFEL. (From *The Brown Boveri Review*, Vol. 28, Nos. 8-9, Aug./Sept., 1941, pp. 232-236).

Introduction.

DIFFICULTIES are encountered in the design of multi-stage turbines and compressors, due to the peculiar characteristics of real gases and vapours. Only in the case of a hypothetical perfect gas with a constant specific heat c_p, is it possible to determine the change in state by purely analytical methods. Already with a semi-perfect gas, notwithstanding the fact that it obeys the relation pv=RT, the specific heat is a function of the temperature, and the assumption that it is constant is justified only if the pressure ratio is small.

In the following, a relation is, therefore, derived by means of which the end point of a polytropic expansion curve may, even in the case of vapours, be determined by a simple process. For perfect gases the method yields mathematically exact results, for vapours, a close degree of approximation is attained. In all cases, the change in condition is determined with an accuracy sufficient for all practical purposes, and so quickly as to make its general adoption very desirable.

Definition of the Polytrope.

It is necessary to explain first what is to be understood by the expressions polytrope and polytropic efficiency as used in the present article, because a polytropic variation of state is not always defined in the same way by different authors.

The polytropic efficiencies η_E and η_K, for expansion and compression respectively, are here used to denote the ultimate stage efficiency of a thermally insulated machine when the number of stages becomes infinitely large, i.e., the efficiency of a thermally insulated stage working between infinitely close pressure limits. Thus in the case of expansion

$$\eta_E = \frac{-di}{-A\,v\,dp}$$

and in the case of compression

$$\eta_K = \frac{A\,v\,dp}{di}$$

In practice, the polytropic efficiency does not differ to any appreciable extent from the efficiency of a real stage η_{st} of finite but small pressure difference, and the turbine designer, for instance, may take $\eta_{st} \backsim \eta_E$ without introducing any error.

By polytrope, we understand a variation of state during which the ratio of di to Av dp remains constant, or

$$\frac{di}{Av\,dp} = \eta_E = \frac{1}{\eta_K} = \text{constant} \qquad (1)$$

The variation of state inside a multi-stage thermally insulated turbo-machine is, therefore, by definition polytropic if all the stages operate with the same efficiency.

The Entropy Increase of the Polytrope.

From the first and second laws of thermodynamics we have

$$dq = di - Av\,dp,$$

and

$$ds = \frac{di}{T} - A\,\frac{v}{T}\,dp \qquad .. \quad (2)$$

Inserting the value of di given by equation (1) yields

$$ds = \frac{1}{T}\,\eta_E\,Av\,dp - A\,\frac{v}{T}\,dp = -(1-\eta_E)\,A\,\frac{v}{T}\,dp \quad (3)$$

This relation says simply that the increase in entropy, ds, is equal to the loss of available work $-(1-\eta_E)$ Av dp, divided by the temperature T.

Further it is readily seen from equation (2)

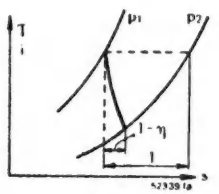


Fig. 1

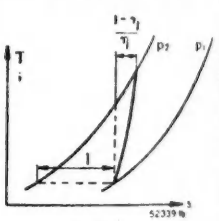


Fig. 2

Fig. 1. Expansion polytrope in the entropy diagram for gases.

Fig. 2. Compression polytrope in the entropy diagram for gases.

The diagrams show the simple relationship between the polytropic efficiency and the real increase in entropy expressed as a fraction of the entropy difference along the line i=constant.

that for i=constant

$$-A\frac{v}{T}dp=\left(\frac{\delta s}{\delta p}\right)_i dp \qquad .. \quad (4)$$

and hence equation (3) may be written

$$ds=-(1-\eta_E)A\frac{v}{T}dp=(1-\eta_E)\left(\frac{\delta s}{\delta p}\right)_i dp.. \quad (3a)$$

So far the treatment has involved applications of the fundamental laws only, so that expression (3a) gives a generally valid relation connecting the increase in entropy and the polytropic efficiency. It applies, therefore, to both gases and vapours.

The Polytrope for Gases.

For gases the quantity v/T, and hence also $\left(\frac{\delta s}{\delta p}\right)_i$ depends only on the pressure and it can, therefore, be directly integrated between the limits of any initial pressure p_1 and final pressure p_2

$$\Delta s=(1-\eta_E)\,A\,R\log_e\frac{p_1}{p_2} \qquad .. \qquad .. \quad (5a)$$

$$=(1-\eta_E)\,(\Delta s)_i, \text{ for expansion,}$$

$$\Delta s=\frac{1-\eta_K}{\eta_K}\,A\,R\log_e\frac{p_2}{p_1} \qquad .. \qquad .. \quad (5b)$$

$$=-\frac{1-\eta_K}{\eta_K}(\Delta s)_i \text{ for compression}$$

In the above, $(\Delta s)_1 = A\,R\log_e\frac{p_1}{p_2}$ is the difference in entropy between the isobars p_1 and p_2 at constant entalphy i. It should be noted that in the case of gases, this relation is strictly true, that is to say, it involves no assumption as to the dependance of the specific heat on the temperature.

Figs. 1 and 2 show how to find in the T-s diagram the end point of a polytrope on any particular isobar, or conversely, how the polytropic efficiency may be calculated from the test data of any turbo-machinery.

The Polytrope in the i-s Diagram for Vapours.

In the case of gases which are not perfect, i.e., in the case of vapours, equation (3a) cannot be integrated directly. In practice, however, the relations (5a) and (5b) can still be used if $(\Delta s)_i$ is replaced by $(\overline{\Delta s})_i$ which is thereby defined as the mean entropy difference of the isobars p_1 and p_2 at constant entalphy i. This $(\overline{\Delta s})_i$ is measured approximately in the middle of the condition curve in the manner shown by the example illustrated in Fig. 3. To be strictly accurate, all the elementary lengths denoted by $\left(\frac{\delta s}{\delta p}\right)_i \Delta p$ along the condition curve should be added together: but a trial will show that the measurement of $(\overline{\Delta s})_i$ in one place is quite accurate enough, no appreciable correction being found if the curve is subdivided into two or even three parts. Only when the condition curve crosses the saturation line does such a subdivision become advisable.

This proves once again the great value of the i-s chart for turbine calculations, for not only does it enable the total adiabatic efficiency to be determined in the usual manner from the ordinates of the initial and final points, but it also makes possible the direct calculation of the polytropic efficiency from their abscissae.

The Polytrope-Entropy Chart for Gases.

In the case of gases it is possible to construct an entropy monogram in which all polytropes (including, therefore, isobars and isothermals) become straight lines and which accordingly may appropriately be called a polytrope-entropy chart, or simply a polytrope chart.

Fig. 4 shows an excerpt from such a chart in which all quantities are referred to 1 mol. The entropy is plotted as abscissae and the quantity $AR\log\frac{p}{p_o}+s$, as ordinates. The latter quantity is nothing other than the entropy s_p along the isobar p_o, that is, $\int_{To}^{T} c_p\frac{dT}{T}$. If the angle α which the

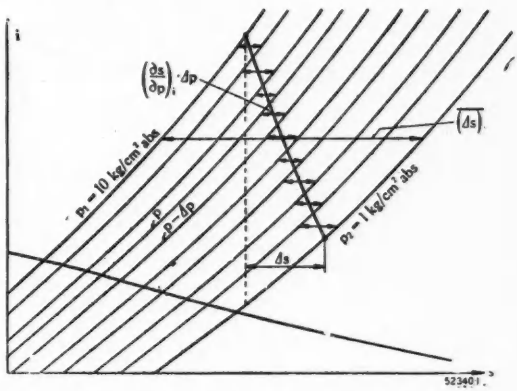


Fig. 3. Construction of the polytrope in the i—s diagram for steam.

The entropy increase Δ_s for an expansion in the i—s diagram for vapours is given by the product $(1-\eta_E)$ into the average horizontal distance $(\overline{\Delta s})_i$ of the neighbouring isobars.

$$\Delta s=(1-\eta_E)\cdot(\overline{\Delta s})_i \qquad (\overline{\Delta s})_i=\Sigma\left(\frac{\delta s}{\delta p}\right)_i \Delta p$$

adiabatic makes with the expansion polytrope at any particular point is calculated,

$$\tan\alpha = \frac{ds}{-d\left(AR\log\frac{p}{p_0}+s\right)} = \frac{ds}{-AR\frac{dp}{p}-ds} = \frac{1}{-\frac{AR}{p}\cdot\frac{dp}{ds}-1} = \frac{1}{-A\frac{v}{T}\frac{dp}{ds}-1}$$

and combining with equation (3)

$$\tan\alpha = \frac{1}{\frac{1}{1-\eta_E}-1} = \frac{1-\eta_E}{\eta_E} \quad .. \quad (6a)$$

and similarly for compression

$$\tan\beta = 1-\eta_K \quad .. \quad .. \quad (6b)$$

Hence for η_E (or η_K) = constant, α (or β) is also constant. This proves that in the proposed diagram, polytropes are indeed straight lines.

During throttling, $\eta_E = 0$, hence $\alpha = 90°$, that is, lines of constant i and isothermals are parallel to the abscissae axis. The values assigned to them will depend on the particular characteristics of the gases concerned: on the two sides of the diagram proper there may be drawn curves of temperature and heat content for any gas. If the diagram is used for one particular gas, the isothermals and the lines of constant i may be drawn directly in the diagram.

The family of isobars is the same for all gases. Its spacing is purely logarithmic, and hence intermediate pressures may be interpolated with the aid of a slide rule scale.

A comparison of the proposed entropy chart for gases with the T-s diagram for gases shows the following differences in the T-s diagram: adiabatics, isobars and polytropes are in general curved. Whilst the family of isobars and,—contrary to the polytrope chart—also the constant volume lines may be used for all gases, adiabatics are differently curved according to the gas. Only in the case of one particular gas, for instance, air, is it possible to ensure that the adiabatics shall be straight. In the polytrope chart on the other hand, all adiabatics, isobars and polytropes are straight lines and remain the same for all gases.

The polytrope chart, therefore, appears particularly well suited to the study of phenomena where a steady flow is maintained, because in such processes, the changes usually take place along polytropes (in special cases along adiabatics, isobars and throttling lines). Where on the other hand, the changes take place mainly at constant volume such as in explosion processes, the T-s chart might be more convenient. It would, of course, be possible to extend the utility of the polytrope chart by the addition of curves of constant volume and curves of u, but in most cases it is sufficient that v may be obtained from RT/p, and u from i-ART.

In regard to the accuracy with which the characteristic quantities of state may be read off on the diagram, this decreases very rapidly in the T-s chart at low temperatures. Such a disadvantage is avoided by the logarithmic scale of the polytrope chart.

Fig. 4 shows by an example how the expansion and compression polytropes are constructed. It is interesting to note that the compression polytropes intersect an isothermal so as to form a linear efficiency scale, the expansion polytropes

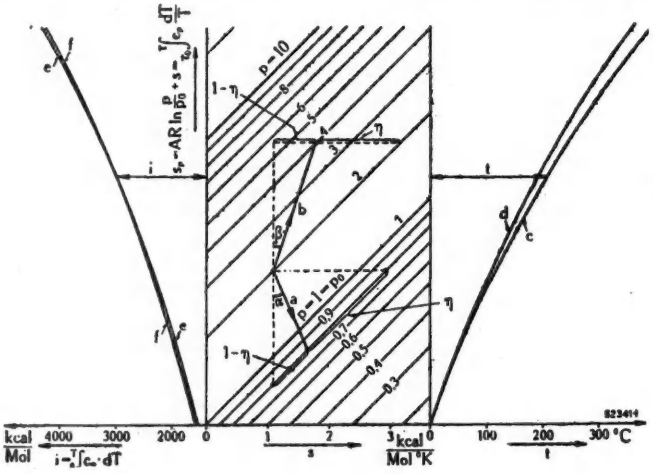


Fig. 4.

Polytrope-entropy chart for gases.

i Entalphy. t Temperature in °C
s Entropy.

Curves:

a. Expansion polytrope
b. Compression polytrope
c. Temperature curve for air
d. Temperature curve for gas-oil combustion gases
e. Entalphy curve for air
f. Entalphy curve for gas-oil combustion gases.

forming a similar scale on an isobar. In other words: if starting from given initial conditions the power of the compressor is kept constant, the increase in entropy is exactly proportional to $1 - \eta_K$, it is a maximum when $\eta_K = 0$ for the initial isobar, and is zero for the adiabatic. In the case of the turbine the relation takes somewhat different form: if, again starting from given initial conditions, the back pressure is kept constant, the increase in entropy is again proportional to $1-\eta_E$, it is a maximum for pure throttling and zero for the adabatic.

It is evident that in principle it would be sufficient to plot only i and t as functions of $AR \log_e \frac{p}{p_o} + s \left(= \int_{T_o}^{T} c_p \frac{dT}{T}\right)$. The drawing of the purely logarithmically spaced isobar lines could be omitted, as the equivalent calculation could readily be carried out on the slide rule, although such a process would not be so illustrative. A compression from a pressure p_1 at a temperature t_1 to a pressure p_2 would then require the following procedure:

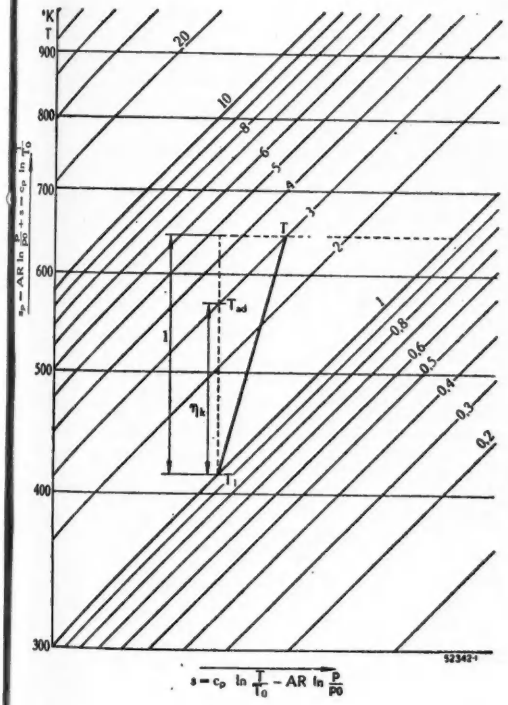


Fig. 5. Compression polytrope for gases with constant specific heat c_p.

With t_1 find on t-curve $AR \log_e \frac{p_1}{p_0} + s_1$.

To this value there must be added

$$AR \log_e \frac{p_2}{p_1} + \Delta s = AR \log_e \frac{p_2}{p_1} + \frac{1-\eta_K}{\eta_K} AR \log_e \frac{p_2}{p_1}$$

$$= \frac{1}{\eta_K} AR \log_e \frac{p_2}{p_1} = AR \log_e \left(\frac{p_2}{p_1}\right)^{\frac{1}{\eta_K}}$$

and obtain thus $AR \log_e \frac{p_2}{p_o} + s_2$ from which the corresponding value of t_2 may be read off.

Polytropes for Perfect Gases with Constant Specific Heat.

If a polytrope chart were to be constructed for a perfect gas with constant specific heat, the temperature scale would be a pure logarithmic one because

$$AR \log \frac{p}{p_0} + s = \int_{T_o}^{T} c_p \frac{dT}{T} = c_p \log_e \frac{T}{T_0}$$

If in such a chart (Fig. 5) we were to draw, for instance, a compression polytrope between the pressures p_1 and p_2, it is easily seen that there is a nice relation between the initial temperature T_1, the real final temperature T and the adiabatic final temperature T_{ad}. Comparing Fig. 4 with Fig. 5.

$$\frac{c_p \log_e \frac{T}{T_1}}{c_p \log_e \frac{T_{ad}}{T_1}} = \frac{1}{\eta_K} \text{ or } \frac{T}{T_1} = \left(\frac{T_{ad}}{T_1}\right)^{\frac{1}{\eta_K}} \quad .. \quad (7a)$$

And similarly for expansion

$$\frac{T}{T_1} = \left(\frac{T_{ad}}{T_1}\right)^{\eta_K} \quad .. \quad (7b)$$

These relations are very convenient in some calculations. They enable, for instance, when making comparisons, to calculate cyclic processes by the temperatures only, without even having to make any assumption as to the value of the adiabatic exponent k. The pressures do not appear at all. Instead of them only adiabatic temperature ratios are used. These can, of course, immediately be converted into pressure ratios by the insertion of a suitable k.

List of Symbols.

Quantities:

p = absolute pressure
v = specific volume
T = Absolute temperature
t = temperature °C

u = internal energy
i = entalphy
s = entropy
c_p = specific heat at const. pressure
k = adiabatic exponent
R = gas constant
q = heat supplied
A = heat equivalent
η = efficiency
α, β = angles between the adiabatic and the expansion polytrope or the compression polytrope respectively in the polytrope-entropy chart.

Suffixes:

E = expansion polytrope
K = compression polytrope
st = turbo-machine stage
0 = general reference state
1 = state before turbo-machine
2 = state after turbo-machine
ad = theoretical final state on adiabatic

AN AUTOMATICALLY RECLOSING BREAKER FOR CO-ORDINATING DISTRIBUTION SYSTEMS

By H. L. RAWLINS and J. M. WALLACE, Westinghouse and Electric and Manufacturing Company.
(From *Electrical Engineering Transactions*, Vol. 60, December, 1941, pp. 1021-24).

IN recent years there has been a large increase in the number of distribution systems having long lines and small loads scattered over large areas. Inexpensive lines with simple fuse protection for faults instead of conventional apparatus for fault protection will provide low installation costs, but result in high maintenance costs and unsatisfactory service. However, more than 80% of all faults are temporary and will not prevent immediate restoration of service. Therefore, the elimination of service trips on such faults will materially reduce maintenance costs. An inexpensive breaker that will interrupt all faults and then restore service after all temporary faults will be most suitable for systems as characterised above.

The fault currents on these systems are relatively low and of such a value that distribution cutouts are used sectionalizing, even to the use of five or more cutouts in series. The excellent co-ordination possible with modern fuses permits definite restriction of outages to the immediate neighborhood of a fault. Though outages on such systems are of long duration, their restriction to limited areas greatly increases the quality of service. Therefore, a requirement for a device to provide fault protection on such systems is its adaptability to an extensive co-ordination scheme. This will be accomplished if the device has a time current characteristic approximating that of modern fuses. The extension of some of the systems concerned makes simple fuses inadequate for sectionalizing duty. Repeating fuses reduce the necessity for immediate service trips, but require a periodic servicing and may not

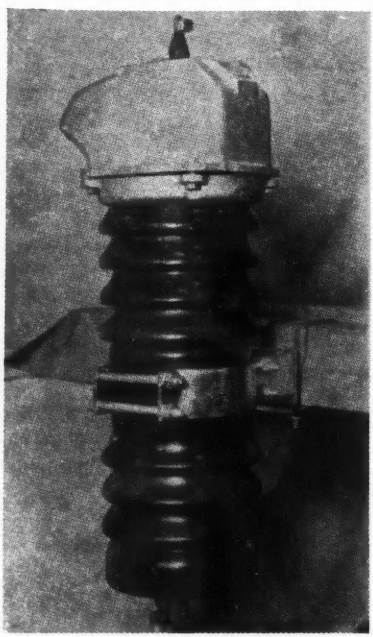


Fig. 1. Single-pole automatically reclosing circuit-breaker.

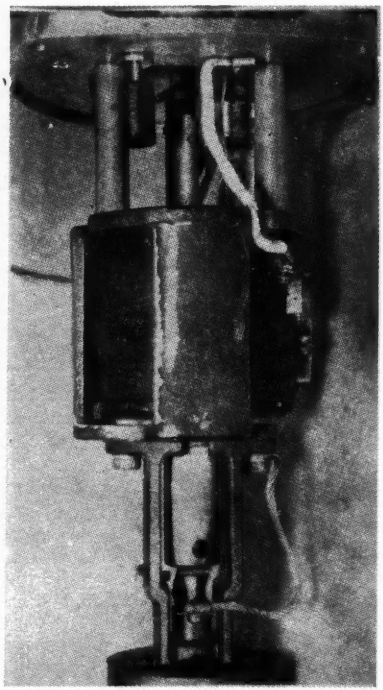


Fig. 2. Interrupter and operating mechanism assembly withdrawn from porcelain tank.

differentiate between a permanent and a temporary fault due to their short reclosing time. Close co-ordination is not dependable with repeating fuses, for the heat stored in a protected fuse may result in its melting upon repeated operation of the fuse protecting it. Therefore, an additional requirement is a time current characteristic, which is not affected by repeated operations of the preceding protective device. From the application view point the most important of the requirements mentioned above is the inverse time current characteristic.

Construction and Operation of the New Breaker.

The single pole automatically reclosing circuit breaker described in this article (Figs. 1 and 2) fulfils the above requirements. Furthermore, it has the following features: trip free closing, immediate manual reclosing after lockout by an indicating handle placed under a protecting hood, lockout on first interruption when manually closed against a fault and multitap coil to permit ready change of rating.

Fig. 3 is a cross sectional view showing all principal elements of the breaker. The breaker is normally biased closed by the contact pressure springs, except during lockout. This differs from more conventional breakers in that, on tripping, the contacts are forced apart against a spring pressure instead of being opened by accelerating springs. The energy required to operate the breaker is obtained from a series coil. The small size of coil used to operate the breaker and continuously carry overload currents just below the minimum tripping current, is made possible by using an efficient type of interrupter that requires only a short travel. Wide separation of the contacts on lockout is obtained by releasing a spring that opens the breaker in the convent onal manner and is reset, on closing, by means of the handle. Additional energy to interrupt heavier currents is obtained from the interrupter itself after the initial separation of the contacts.

The armature when actuated by an overcurrent through the service coil moves upward and forces the entrapped oil from the timing dashpot. This action provides the inverse time characteristic on tripping, shape of which depends on dashpot size and port area together with the magnetic circuit. After a predetermined travel, the dashpot is opened, permitting the free armature to drive the moving contact open against the contact pressure springs. The moving contact opens rapidly for a short distance and then picks up the orifice piston, which forces oil into the arc path resulting in interruption at the end of the first half cycle after the orifice piston starts to travel. On high currents the pressure developed within the interrupter will produce an additional driving force on the plunger, thereby forcing oil into the arc path under a pressure proportional to the value of current to be interrupted. The moving contact is free to move upward through the armature, thus permitting overtravel on high current interruptions. The moving contact is prevented from reclosing at once by the delay latch, which is not released until the armature reaches its starting position. This results in a reclosing time of approximately five seconds. The time interval before reclosing permits the escape of gas from the top of the interrupter and entrance of fresh oil into the bottom. This flushing takes place through washer type valves, that close instantly upon the appearance of pressure during interruption.

There is provided a simple, three step lockout trip which is notched up by the travel of the armature during each interruption. On the third

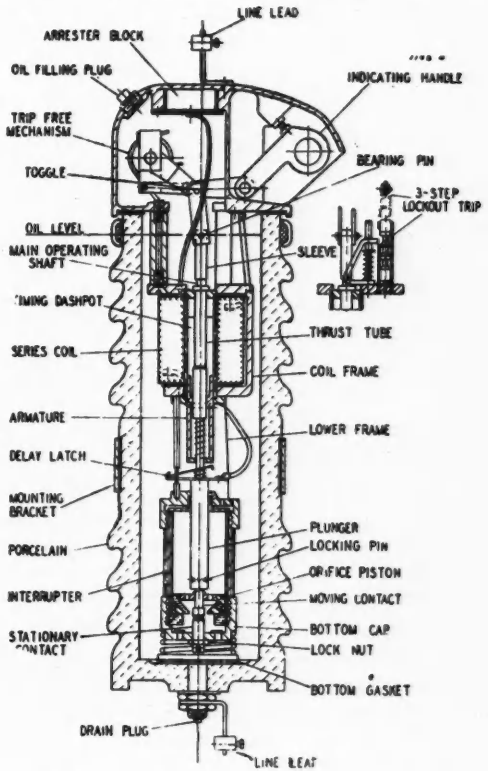


Fig. 3. Sectional view of breaker with side view of lockout trip.

interruption it breaks the toggle that holds the trip free mechanism in the closed position, thereby permitting the contacts to be driven to the extreme position. The manual return of the indicating handle to the closed position will reset this toggle and the lockout trip and permit the breaker to close after the usual time delay. The lockout trip will be reset only to its second position from which it will lock out the breaker if there is a further interruption. If the fault has been removed from the line, the lockout trip will slowly reset to provide three step lockout on subsequent faults.

The series coil is made up of four separate windings. By connecting these windings in series, series-parallel or parallel one obtains six different characteristics corresponding to 5, 10, 15, 20, 30 and 60 ampere. The coil is protected from transient voltages by a suitable arrester connected directly across its terminals.

Together with the indicating handle, an oil gauge and an operation counter is grouped under the hood.

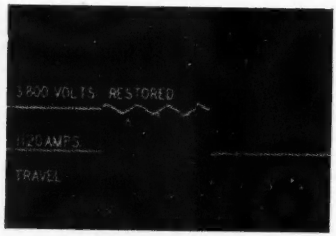


Fig. 4. Interruption of 1120 amperes at 13,800 volts.

Interruption and Lockout Tests.

Repeated tests have demonstrated the high efficiency and low oil deterioration of the breaker. Indications are that the breaker can be in service many years before an oil change is required. Figs. 4 & 5 show some oscillograms of interruptions.

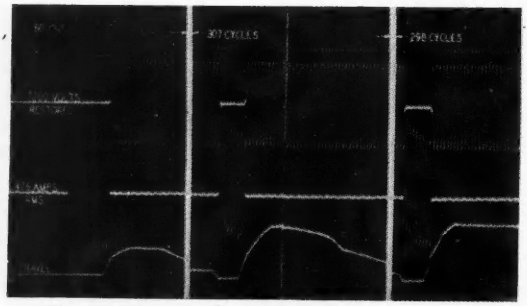


Fig. 5. Three interruptions and lockout, 476 amperes at 7,200 volts.
Note the travel of the contacts to the full open positions after final interruption. (Portions of film showing breaker open between reclosures have been deleted from the original eight-foot film).

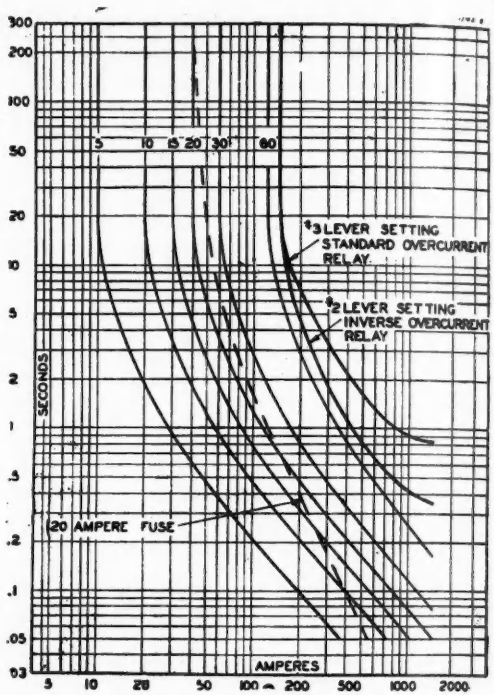


Fig. 6. Tripping-time current characteristics of new breaker.

Co-ordination Characteristics.

The important attribute of this breaker is the time current characteristic. More time delay on tripping allows more time for co-ordination and permits, therefore, closer co-ordination between adjacent ratings. Fig. 6 shows the time current characteristics for the six ratings. All of these ratings can be co-ordinated in series. The possibility of co-ordination between these breakers and fuses is further demonstrated by superimposing a 20 ampere fuse curve upon the breaker curves. The fuse curve is for an universal fuse link as used in distribution cutouts.

The relay curve shown in Fig. 6 is for standard over-current relay of the moving disc type. It is possible to adjust the relay curve close to that of the breaker. With the setting shown the breaker will consistently protect the relay and thus prevent tripping an entire substation. The time delay between operations of the reclosing breaker is sufficient to allow the relay disc to return to its starting point, so the co-ordination will be the same for each operation. This would not be the case with reclosing fuses. Here the relay disc would not return to its original position due to the short reclosing time of the fuse.

A further advantage of the breaker lies in the economies offered by the use of single gaps for lightning arresters in low capacity systems. On these systems a temporary outage is not serious, and with the co-ordination afforded by these breakers such an outage due to lightning is confined to a small area and is removed upon first reclosure.

RADIOGRAPHY AT HIGH SPEED

By LOUIS F. EHRKE and CHARLES M. SLACK. (From *Electrical Engineering*, Vol. 60, No. 9, September, 1941, pp. 432-35).

INTERESTING results have been attained by ultra high speed photography using X-radiation instead of visible light, making possible to record distortion or change in the interior of opaque objects. To make such radiographs at speeds of about one micro-second, enormous currents must be passed through the X-ray tube for that very short time interval. In the conventional hot-cathode X-ray tube, the available electron current is limited by safe cathode temperature and space charge effects. These limitations are avoided by the use of a cold-cathode X-ray tube recently developed in the Westinghouse lamp research laboratory at Bloomfield, N.Y.

This tube employs a pair of cold electrodes G and H (Fig. 1). G is the true cathode. The third electrode H is placed close to G and connected to the anode through a high resistance (Fig. 2).* This electrode concentrates a high field at the cathode and initiates the discharge, which is then forced, by the voltage drop across the high resistance J to transfer to the anode. As soon as this auxiliary electrode has performed its job of starting the discharge, it assumes a potential very close to that of the cathode and consequently if shaped correctly (as in Fig. 2 for instance) may be utilized in focusing the electrons on the desired region of the cathode. Currents of from 1000 to 2000 amperes pass through the tube for periods of about one micro-second. The X-ray energy produced is sufficient to produce radiographs of rapidly moving objects without blur caused by motion.

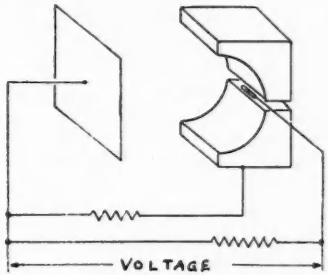


Fig. 2. Arrangement of anode and auxiliary anode.

A basic circuit is shown in Fig. 1. The high voltage transformer A (125 kV, 5 milliamperes) feeds capacitors L through rectifiers M. The capacitors (60 kV, 0.04 microfarad, two only needed for most applications) are charged to a potential slightly less than that required to break down the gap F. When it is desired to fire the tube by passing the capacitive charge through it, the firing circuit is opened at B. This permits

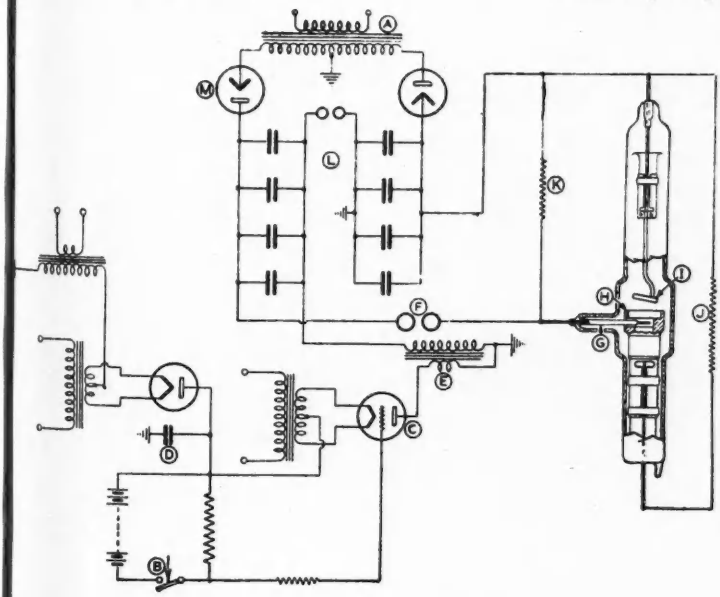
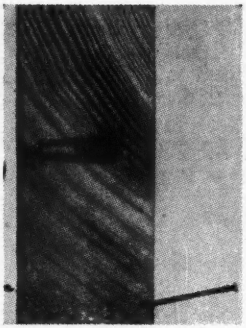


Fig. 1. Circuit diagram of the new ultra high-speed X-ray tube and auxiliary equipment.

*This figure is from a publication of the same authors: Slack and Ehrke, Field Emission X-ray Tube, *Journal of Applied Physics*, Vol. 12, No. 2, February, 1941, pp. 165-68.

(a) Bullet entering block.



(b) Bullet has entered block; note rapid closing of hole by wood fibres.

(c) Bullet emerging from block.

(d) Bullet has left block. Note lead particles broken from bullet in its passage.

Fig. 3. Radiographs showing penetration of a projectile into a wooden block.

the negative charge on the grid of the thyratron C to leak off, and the charge on capacitor D then passes through the thyratron and through the induction coil E to ground. This sudden passage of current produces a high voltage in the secondary of E, which raises the potential of capacitor bank L so that the total voltage across the capacitor is now great enough to break down the gap F. When this occurs, the high voltage is placed between G and H and starts a discharge between these electrodes. By the blocking action of the resistance J (100.000 ohm.) the electron stream from the cathode " hotspot " on G is diverted to the anode I, directed and focused by H. X-rays are generated at I by the impact of this electron stream. Resistance K is placed directly across the capacitors and gap to insure that the breakdown at gap F will be consistent with its spacing.

Current measurements indicate that peak currents from 1000 to 2000 amperes are common. The time during which this large current passes is about one micro-second. The method used to measure this extremely short time interval is as follows: A 0.22 Swift bullet was radiographed in free flight. The velocity of this projectile is about 4.100 feet per second. Therefore the projectile moves 1.25 mm. in one micro-second. The amount of blurring in the radiograph, in most cases 1 mm., indicates the duration of the X-ray exposure being about one micro-second.

The usual method of firing the tube is to bridge switch B with a fuse wire which is broken with the projectile, in the case of the bullet pictures. Other arrangements may be used for other objects.

Fig. 4. Radiograph showing a foot striking a football.

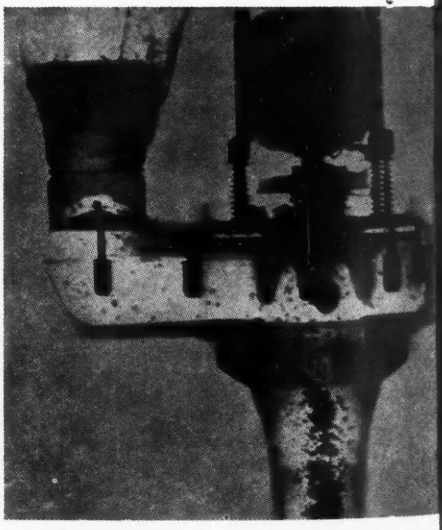


Fig. 5. Radiograph of the interior of a vacuum cleaner

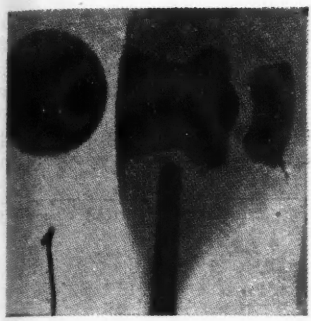
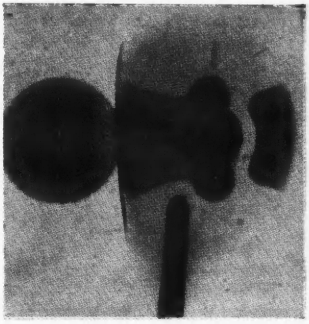
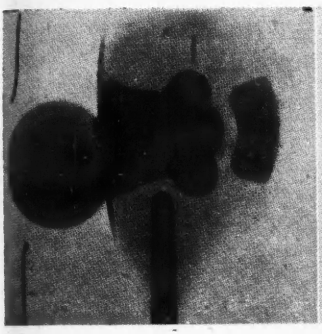


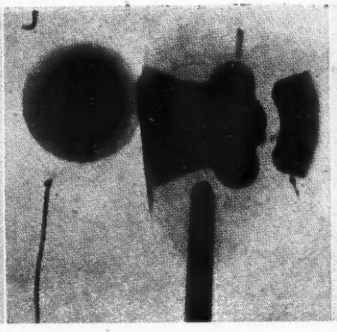
Fig. 6. Radiographs showing club striking golf ball.

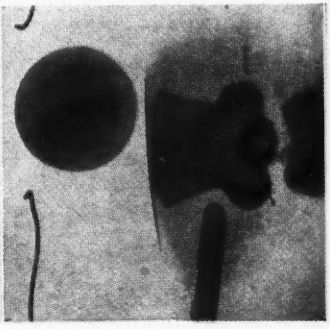
(a) Club head approaching ball.
(b) Core of ball partly compressed.
(c) Maximum compression of core.
(d) Core starts to re-expand.
(e) Ball leaving club.



(a) (b)







(c) (d) (e)

The time interval which elapses between breaking of the contact and the firing of the X-ray tube may be controlled by suitable choice of the grid leak resistor in the thyratron circuit. The methods of firing are adaptable to a wide variety of conditions.

Fig. 3 shows radiographs of the penetration of a projectile into a wooden block.

Fig. 4 is a radiograph showing a foot striking a foot-ball.

The interior of a vacuum cleaner is shown in Fig. 5. Flow and relative air velocity can be judged by the distribution of the particles.

Fig. 6 is a series of radiographs showing club stroking golf ball. Series sets of one object are made up of separate X-ray shots of separate specimens, e.g., similar blocks of hard wood in Fig. 3, the exposure being so spaced through the cycle, that a series of consecutive discrete phases is recorded.

The voltage used to produce the pictures shown here was in the region of about 120 kV peak. The wave length of the X-ray radiation produced at this voltage is about 0.1 to 0.2 angström or about 10^{-9} cm. The X-ray pictures are shadowgraphs of course. The closer the object can be placed to the film, the better the definition obtained. For this reason the bullet pictures made by firing close to the film show better definition than does such a picture as that of the football and foot which are necessarily farther from the film.

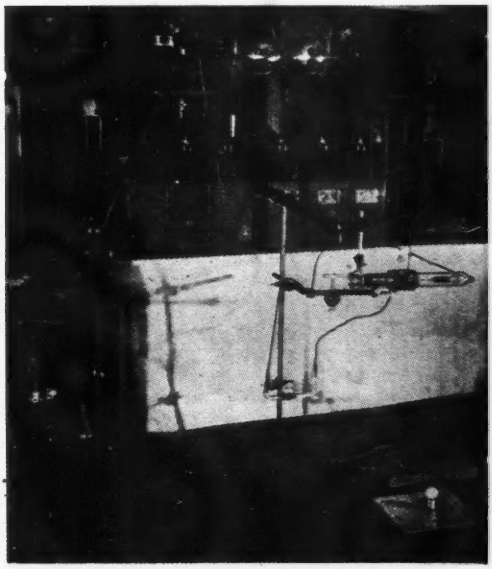


Fig. 7. A typical setup as used for taking a golf picture.

It was possible to penetrate with 100 kV and 0.02 microfarad and using good intensifying screens 4" of tissue, 4 cm. of aluminium and 2 mm. of steel. The penetration goes up rapidly with the voltage, however, and it is believed that with 200 to 300 kV it should be possible to obtain distinguishable radiographs through an inch or more of steel equivalent. This voltage is a great deal higher than that necessary for long time exposures, and consequently there will be considerable loss in contrast. Fortunately this loss of contrast can be tolerated in most commercial applications because of the great differences in opacity of materials, and it is believed that a great deal of valuable information might be obtained in the study of the distortion of rapidly moving machine parts as well as the penetration of bullets in various materials*.

* Content of the last paragraph from Slack and Ehrke, loc, cit.

ELECTRIC TRACTION VEHICLES

(From *Bulletin de la Société d'Encouragement pour l'Industrie Nationale*, Vol. 139/140, July, 1940, to June, 1941, pp. 192/194).

THE Société Centrale de Chemis de Fer et d'entreprises has devoted itself for many years to the development of accumulator driven vehicles, especially since the International Exhibition in Paris, 1937, where it provided the internal transport. For this purpose small electric tractors and trailers and electric taxis were used. These vehicles had been manufactured partly by the Société Centrale and partly by several specialized factories.

Utilizing the extended experience gained through this service and elsewhere the company has now started production in great numbers of commercial and inductrial vehicles. Fig. 1 shows an accumulator driven tractor with a trailer. These vehicles are lent to factories and merchants for deliveries in Paris and its neighbouring districts.

Another type of vehicle, a delivery van, shown in Fig. 2, is manufactured for three different loads: 500, 1000 and 2000 kg. The principal features of this range of vehicles are the patented pressed steel chassis and pressed sheet body, and the front wheel drive through a reduction gear and flexible coupling. The floor of the chassis is only one foot above the ground and no runnery boards are provided, as well as only one side door and one rear door. The cells are arranged in two batteries, one under the motor and one between the rear wheels. The batteries are very accessible and slide out of the vehicle on trays. Only one type of battery is used for all types of vehicle. The object in view was to simplify manufacture and maintenance.

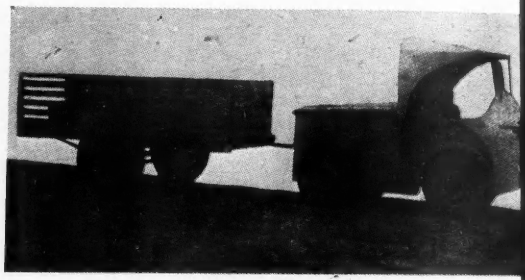


Fig. 1

Fig. 2. Accumulator driven delivery van.

Fig. 1. Accumulator driven tractor with trailer.

Approximate useful capacity of the trailer	7 m³.
Maximum useful load	2000 hg.
Maximum distance to be covered on a good road without recharging the battery	70 m
Speed	25 km/hr.

	Useful load of the van: 500	1000	2000
Nett weight of vehicle and battery	750	900	1100
Length and width of vehicle	3.65×1.44	4.5×1.44	5.3×1
Internal height of vehicle	1.57	1.57	1.65
Useful capacity	4.5	6	10 m

STRESSED SKIN STRUCTURES IN PLYWOOD

By S. BLUMRICH. (From *Luftfahrtforschung*, Vol. 18, No. 19, September, 1941, pp. 331-337).

THE main drawback of the usual shell structures of to-day lies in the fact that the skin plating cannot be fully utilized, either in compression or in shear. In shear, the secondary stresses due to diagonal folds become soon excessive; in compression, after buckling of the skin, the stringers carry an appreciable portion of the load, the more so the higher the load. To increase the buckling load of the skin, the stringers could be placed closer to one another, which design in sheet-metal work leads to uneconomical construction owing to increased riveting costs. Therefore, economical considerations induce the engineer, to-day, to use a thicker skin and less stringers—a solution unfavourable from the weight point of view.

However, by using plywood as skin, this is not the case, and the skin can be made so as to buckle at the same time when the stringers buckle. In this fashion the total cross-section becomes equally efficient, thus reducing the structural weight per unit load carried. This type of construction, called in the following "reinforced plywood" will be compared with the standard type in which the skin and stringers buckle at different stresses.

I. THEORETICAL INVESTIGATIONS

1. Reinforced Plywood.

(a) Flat Panels.—The requirement of equal buckling stress for skin and stringers makes a very small plate width (i.e., very close stringer spacing) necessary, compared with customary dimensions. The same applies, of course, to the cross-sectional area of the stringers. By choosing suitable cross-sections for the frames so as to allow them to deflect, the length of stringer acting as a strut is not limited to the distance of the frames, and the possibility of great variation of all dimensions is open for a given load.

Fig. 1 shows a panel of reinforced plywood. This scheme involves no cutting of stringers or frames. The system skin-stringer is simply glued to the frames, the available area for gluing is 2.a.b. Thus one variable in determining the buckling load of the skin is eliminated, the plate length being taken infinite.

Table I. gives the dimensions and buckling loads under compression of all panels investigated. For the buckling load the validity of H. Wagner's formula ("Sheet-Metal Airplane Construction" in A.S.M.E.-Transactions 1931, AER-53-18, p. 151) has been analysed.

$$P = \frac{20 \cdot E}{B} \cdot \sqrt{I_y^* \cdot I_x^*} + \frac{10 \cdot G}{B} \cdot T_v^*$$

$I_y^* = I_y/l$, $I_x^* = I_x/l$, I_y and I_x are moments of inertia (Fig. 1), E = Modulus of Elasticity.

The second part of the right hand side of the equation containing the torsional stiffness has been neglected. The tabulated value F is the area of one plate plus one stringer. The difference of the moduli of the skin (across the grain) and of pine was taken into account by putting $E_{skin} = 0.7\ E_{pine}$ ($E_{pine} = 120{,}000$ kg/cm^2 $= 1.7 \times 10^6$ lb./sq. in.). The variation of the Moment of Inertia along the panel width B was taken to be

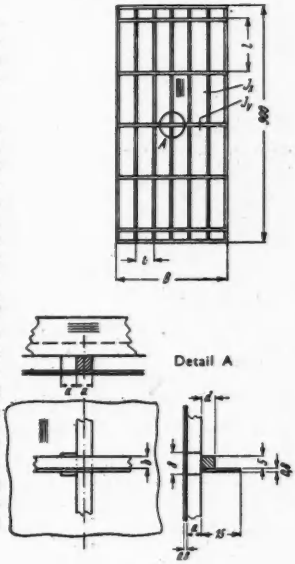


Fig. 1

TABLE I.

a mm.	l mm.	t mm.	No.	F cm²	I_x cm⁴	I_x cm³	i cm.	I_y	I_v	Buckling Calculation.	Load kg. Expt.
		30	1	0.46	0.00868	0.00289	0.138			336	300
	100	50	2	0.66	0.00968	0.00194	0.121	0.2597	0.02597	250	250
		70	3	0.86	0.01028	0.00147	0.109			212	200
		30	4	0.46	0.00868	0.00289	0.138			256	200
4	200	50	5	0.66	0.00968	0.00194	0.121	0.3014	0.01507	191	225
		70	6	0.86	0.01028	0.00147	0.109			162	150
		30	7	0.46	0.00868	0.00289	0.138			216	400
	300	50	8	0.66	0.00968	0.00194	0.121	0.3219	0.01073	161	150
		70	9	0.86	0.01028	0.00147	0.109			136	150
		30	10	0.66	0.03095	0.01032	0.216			744	350
	100	50	11	0.86	0.03655	0.00731	0.206	0.3584	0.03584	572	400
		70	12	1.06	0.03955	0.00571	0.194			489	300
		30	13	0.66	0.03095	0.01032	0.216			577	550
6	200	50	14	0.86	0.03655	0.00731	0.206	0.4254	0.02127	442	300
		70	15	1.06	0.03995	0.00571	0.194			376	350
		30	16	0.66	0.03095	0.01032	0.216			488	500
	300	50	17	0.86	0.03655	0.00731	0.206	0.4549	0.01516	375	250
		70	18	1.06	0.03995	0.00571	0.194			318	250

$$I_x' = \frac{2}{B} \Sigma I_x \cos^2 \frac{\pi y}{B}$$

The column length of stringer was calculated by the formula

$L = 0.91\ B \sqrt[4]{\frac{I_x^*}{I_y^*}}$, and is given in Table II.

TABLE II.
Column-length, cm.

No.	Calc.	Expt.	No.	Calc.	Expt.
1	21.3	—	10	27	30
2	19.2	21.5	11	25	25
3	18.1	—	12	23.4	25
4	24.4	25	13	31	24
5	22.2	20	14	28	25
6	20.7	20	15	27	—
7	26.6	20	16	34	18
8	24.0	25	17	31	25
9	22.5	—	18	29	21

To the derivation of the failing load the denomination is given in Fig. 2.

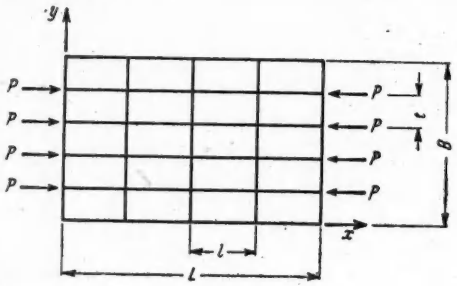


Fig. 2

Fig. 3 indicates the loads on the buckled stringer and on the deflected frame respectively.

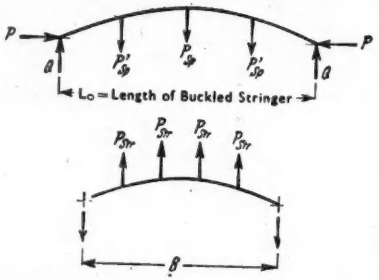


Fig. 3

Accordingly, the stringer represents an elastically supported strut, whereby the supporting forces (P_{SP}) from the frames are put proportional to the deflection of the stringers, i.e., to $\sin \frac{\pi . x}{L_o}$. Then, if P_{SP} is the force at the middle of the stringer

$$P'_{SP} = P_{SP} \sin \frac{\pi . x}{L_o} = P_{SP} \cos \frac{\pi . a}{L_o}.$$

and from Fig. 4

Fig. 4

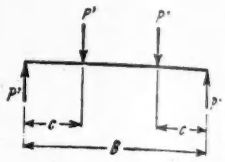


Fig. 5

$$e' = e \sin \frac{\pi . x}{L_o}$$

The maximum bending moment at mid-stringer is

$$M = P . e + \Sigma P'_{SP} . a - Q \frac{L_o}{2} \quad .. \quad .. \quad (1)$$

Hereby $$a = \frac{L_o}{2} - x,$$

and $$\Sigma P'_{SP} . a = P_{SP} \Sigma \cos \frac{\pi . a}{L_o} . a \quad .. \quad .. \quad (2)$$

Moreover $$\Sigma P'_{SP} = P_{SP} . \Sigma \cos \frac{\pi . a}{L_o} = Q \quad .. \quad (3)$$

Hence,

$$M = P . e + P_{SP} \left(\Sigma \cos \frac{\pi . a}{L_o} . a - \frac{L_o}{2} \Sigma \cos \frac{\pi . a}{L_o} \right) \quad (4)$$

In conformity with J. Ratzersdorfer (*Die Knickfestigkeit von Stäben und Stabwerken*, Wien, 1936, Julius Springer) we introduce the value A as the force necessary to a 1 cm. deflection of the frame. Then

$$P_{SP} = A.e. \quad .. \quad .. \quad (5)$$

For a simplified frame loading (Fig. 5) the deflection, f, is :

$$f = \frac{P' . c}{24 E I} (3B^2 - 4c^2)$$

With n = number of plates, m the denomination of the particular station

$$c = B \frac{m}{n}$$

and

$$f = \frac{P'}{24 E I_v} \cdot B \cdot \frac{m}{n} \left[3B^2 - 4B^2 \left(\frac{m}{n} \right)^2 \right]$$

Distributing P' over B according to $\sin \frac{\pi . c}{B}$:

$$f = \frac{P s_{tr} \sin \frac{\pi m}{n} B^3}{24 E I_v} \cdot \frac{m}{n} \cdot \frac{3n^2 - 4m^2}{n^2},$$

P_{str} being the load at the maximum deflection.

Under the influence of all the forces P', Σf is summated, and for $\Sigma f = 1$ cm, one gets $P_{str} = A$, with a factor

$$C = \Sigma \sin \frac{\pi . m}{n} \cdot \frac{m}{n} \cdot \frac{3n^2 - 4m^2}{n^2} \quad .. \quad (7)$$

and the force on the frame

$$A_{SP} = \frac{24 . E I_v}{C B^3} \quad .. \quad .. \quad (8)$$

By substituting equation (8) into equation (5)

$$P_{SP} = \frac{24 E I_v}{C B^3} . e$$

thus the moment equation (4) becomes

$$M = P.e. + e.\frac{24.E.I_y}{C.B^3}\left(\Sigma\cos\frac{\pi.a}{L_o}.a - \frac{L_o}{2}.\Sigma\cos\frac{\pi.a}{L_o}\right)$$

The failing stress of the stringer

$$\sigma_{BR} = \frac{P}{F} + \frac{M}{W} \quad .. \quad .. \quad .. \quad (9)$$

whereby $W = \frac{I_x}{s}$, s being the distance of centroid from the inner layer.

Putting $K = \Sigma\cos\frac{\pi.a}{L_o}a - \frac{L_o}{2}\Sigma\cos\frac{\pi.a}{L_o}$, .. (10)

and transforming equation (9) the buckling load P becomes :

$$P = \frac{F.W}{W+F.e}\left(\sigma_{BR} - 24E\frac{e}{B^3}\,s\,.\,\frac{K}{C}.\frac{I_y}{I_x}\right) \quad .. \quad (11)$$

Results are given in Table III.

TABLE III.

l mm.	t mm.	Breaking force P kg Calculation	Experiment
100	30	75	73.5
	50	112	95.5
	70	141	113.0
200	30	67	74
	50	97	99
	70	121	125
300	30	56	78
	50	78	86
	0	97	115

(b) Cylindrical Shells.—Here the frames are so dimensioned (Fig. 6) as to be rigid, and thus the calculation is simplified by taking the element stringer plus skin as a column buckling between two frames. The M.I. of the shell cross-section is taken as that of a circular ring with a mean thickness of

$$\delta_m = \delta + \frac{a^2}{t}$$

The buckling values have been calculated for stringer plus full effective width t.

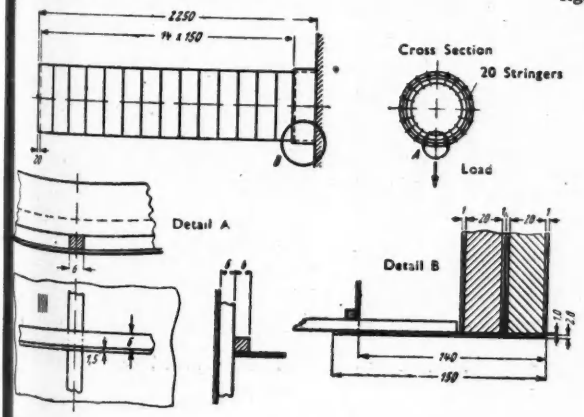


Fig. 6

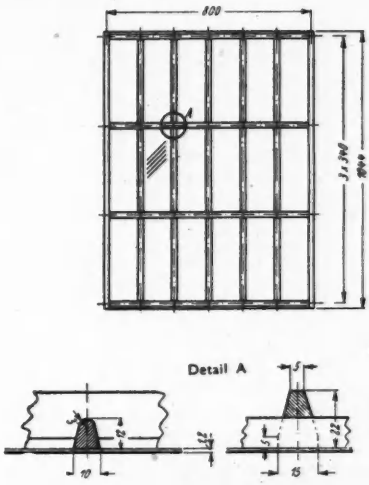


Fig. 7. Panel of standard design.

For the shell in Fig. 8 the values were :

$I = 9500$ cm⁴, radius of gyration $i = 0.187$ cm.

$W = 346$ cm³, and slenderness ratio $\lambda = 80$.

The calculated buckling stress was $\sigma = 173$ kg/cm²

The column-curve used is given in the appendix.

2. Standard Construction.

(a) Flat Panels.—In the standard type of panel the frames are rigid, an assumption which was confirmed in subsequent tests.

The section properties of one stringer plus the effective width of 3 cm are :

$F = 1.02$ cm², $I_x = 0.197$ cm⁴, $i = 0.44$ cm.

With a frame distance of 340 mm. λ becomes 78, and the buckling stress (from column-curve in appendix) $\sigma_K = 175$ kg/cm².

As the grain of the skin plating runs diagonal, the E-modul was taken 40% of that of pine. The panel is shown in Fig. 7.

(b) Circular Shells of Standard Type.—The design can be seen from Fig. 8. The frames are, of course, rigid again.

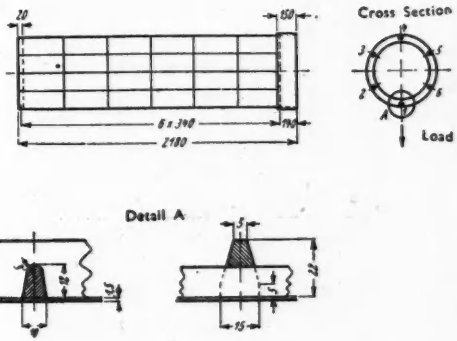


Fig. 8. Shell of standard design

Values for the total cross-section including the skin are :

$I = 11760\ cm^4$, $W_{min} = 428\ cm^3$,
$W_{12} = 496\ cm^3$ (for axis through mid-plate).

As the influence of plate width on the buckling strength of curved plywood plates is not yet known we had to revert to experiments. The moment at buckling of the plates has been found (see chapter II., 2b) to be 46800 cm.kg. Keeping this value constant during a further increase in loading means then that the stringers must carry the increase of moment up to their failure. The stringer stress at buckling of the plate is

$$\sigma s_{tr} = \frac{46{,}800}{W_{min}} = 109\ kg/cm^2$$

The buckling stress of the stringer plus the effective width of 4 cm. longitudinal-grain skin is 175 kg/cm². The difference in stress of 66 kg/cm² is in accordance with the additional moment necessary to bring about failure. The M.I. of the stringer alone is

$$Is_{tr} = 2230\ cm^4$$

For the outermost stringer-centroid $W = 82.5\ cm^3$, and thus $\Delta M = 5450$ cm.kg.

The estimated failing moment is then

$$M_{BR} = 46{,}800 + 5450 = 52{,}250\ cm.kg.$$

which corresponds to a load of

$$P = 282\ kg.$$

II. EXPERIMENTS.

1. Reinforced Plywood.

(a) Flat Panels.—To achieve even load distribution each stringer was loaded separately, and the transverse edge members of the panel consisted of two closely spaced frames (Fig. 1).

It has been maintained that the available area for gluing of stringers to frame is 2.a.b. Obviously, at large stringer deflections when the tensile strength of the glue is exceeded, this joint ceases. An immediate increase in deflection up to Euler failure of the stringer follows. With the exception of panel No. 9 all failures of the ply skin were due to buckling of the inward grain. In panel No. 9 the skin failed in tension.

No failure was registered with the frames, in some cases their plywood web buckled only.

(b) Cylindrical Shells.—Fig. 6 shows the shell, and detail B its fixing. The load producing the bending moment is also shown. Fig. 9 gives a photo. of the buckled shell, and it can be seen that the stringers are unbuckled. This means that they buckle at a higher stress than the ply skin, whereas it was intended to let both buckle simultaneously. The reason for this discrepancy lies with the unknown influence of the plate width on the buckling stress, but, also, in order to get direct comparison with the standard construction, the two failing loads had to be kept nearly constant, which influences the plate width too.

The total deflection at $P = 275$ kg was 23 mm. Failure occurred at $P = 320$ kg, corresponding to a moment $M = 64{,}600$ cm kg. With $W = 346\ cm^3$ (see Chapter I., 1b) the failing stress is $\sigma = 187\ kg/cm^2$. The estimated value was $\sigma = 173\ kg/cm^2$.

Fig. 9. Reinforced plywood shell under bending load=250 kg.

Fig. 10. Shell of standard design under bending load=250 kg.

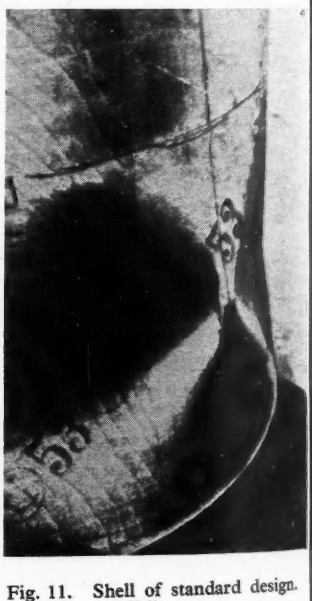


Fig. 11. Shell of standard design. Failure.

2. Standard Construction.

(a) Flat Panels.—In these tests the assumption of the rigidity of frames was confirmed, and the stringers found buckled between the rigid frames only. The average failing load of six tests was 181 kg, which gives with an area F = 1.02 cm², a failing stress of σ = 178 kg/cm². The calculated value was σ = 175 kg/cm² (see Chapter I., 2a).

(b) Cylindrical Shell.—The layout is given in Fig. 8, the fixing of the enclosed end is analogous to that of Fig. 6. At the load P = 250 kg, some plates buckled (see Fig. 10).

The moment was M = 187 . 250 = 46,800 cm kg. With W_{12} = 496 cm³, the buckling stress of the ply skin is σ = 94 kg/cm². As in the case II, 1b, the deflection has been increased here too, and was 32 mm. at a load of P = 275 kg. In spite of the greater wall thickness the deflection was much greater here. This can be explained by the rapid decrease of the shear modulus of plywood. The stringers started to buckle at P = 280 kg, and tears appeared in the ply skin. The stringer failed at P = 295 kg., whereby the tearing of the skin increased rapidly. A photograph of the shell in failure is given in Fig. 11.

III. WEIGHT COMPARISON.

The weight comparison is based wholly on the experimental data, in comparing the reciprocal values of the stresses, i.e., the areas per maximum loads carried.

For flat panels of reinforced plywood the average value from all tests was 0.0096 cm²/kg, against 0.0137 cm²/kg, the average for the standard construction. In the comparison of flat panels the frame weights are not included.

For the comparison of the shells, the actual weights were G = 5.37 kg to 1957 mm. length, for reinforced plywood, and G = 6.47 kg to 2004 mm. length for the standard type. Taking into consideration the respective moments at failure, the weight ratio is then :

$$\frac{G_{\text{reinforced plywood}}}{G_{\text{standard construction}}} = \frac{5.37}{1957} \cdot \frac{2004}{6.47} \cdot \frac{55{,}000}{64{,}600} = 0.723$$

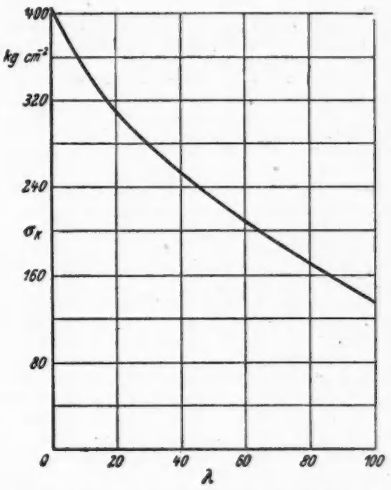


Fig. 12. Column-curve for pine-wood

The investigation shows that with the reinforced plywood design a weight saving of 5 to 50% can be effected.

IV. APPENDIX.

Owing to the great scatter in strength characteristics a theoretical derivation of the column-curve for wood is rather doubtful. An attempt has been made by Roš and Brunner, but this is for timber. For selected species of wood such as employed in aeroplane structures, no column-curves have been published as yet. Therefore, tests were carried out, the result of which is given in Fig. 12. This has been used throughout the present work.

ELECTRIC ELEVATORS IN INDUSTRIAL PREMISES WITH EXPLOSIVE ATMOSPHERES

By Dipl. Ing. O. Sander. (From *Die Wärme*, Vol. 64, No. 48, November, 1941, pp. 439-445).

Prior to planning a lift installation it must be decided whether or not the building to be served has an atmosphere liable to explode or directly connected with such rooms or workshops. Not only the rooms where materials which generate explosive gases are handled are to be regarded as such, but also the space connected with these by means of doors, windows, ducts, etc., if there is even a remote possibility of explosive gases escaping from the actual workshop. Such is the case with oil refineries, Buna, artificial silk producing plants, gas works, etc. Even certain mixtures of solid particles with air (e.g. coal dust) can form an explosive mixture.

Apart from the possibility of spontaneous combustion of many of these atmospheres explosion may occur as a result of electrical sparks, sparks caused by machinery (impact) or by overheating of a part of the machinery in contact with the explosive mixture. If the workshop is not completely enclosed, that is, there is a plentiful flow of air, the danger of an explosion is considerably reduced and is usually located to the immediate vicinity of the machinery or other installation where the explosive mixture is actually formed. But even in such instances dangerous "pockets" may be formed and artificial ventilation could very seldomly, and under special circumstances only, be regarded as an adequate protection against explosion.

Danger of Explosion through Lift Installation.

It is obvious that when a lift is installed in a building where an explosion might occur, all parts of the lift must be prevented from sparking and from overheating.

Regarding explosion caused by electric sparks, numerous experiments were carried out to throw light upon the occurrence during ignition. From these tests it was found that a critical voltage or current density that would cause explosion could not be fixed. Many other factors are of great importance too, such as A.C. or D.C., the arrangement of the traction machinery, etc. Also the impedances in the circuit, independently of the power, whether capacitive or inductive are important factors in sparking. The shape and material of electrodes, too, have to be considered as well as the duration of the spark. Whether it is a thermal or an electrical process or a combination of both which leads to the explosion is irrelevant from the point of view of this investigation. As an example, according to Dr. Ing. Muller-Hillebrand, a short

circuit of 100 V. and 1.0 A. will explode methane and one of 0.34 A. sulphur carbonate. Similarly, an explosion may be caused by the sparking of an electric bell of 4.5 volt. and 1.0 A., when the inductance is 0.35 mH. According to other investigations sparks in circuits of 6 V. and 250 mA. and 0.1 H. may ignite a coal gas and air mixture. A methane and air mixture may be ignited by sparks of 10 V., 350 mA. and 0.1 H. inductance. Dr. Ing. Zas states that light petrol or benzol and air mixtures may be ignited by the sparking of a 12 volt. battery. Although there are no comprehensive results of the experiments carried out on this line of investigation it can be generally said that sparks resulting from currents of 1 A. may cause an explosion when the voltage is only a few volt.

Now the electric power supply to elevator motors is usually 220 or 110 volts., and sometimes as low as 60-12 volts. The current density of auxiliary equipment such as doors, relays, etc., is, on the other hand, usually more than 0.5 A. and the inductance of the circuits higher than 0.35 mH. It is obvious, therefore, that these values are much greater than those which initiate explosions, and in general elevators are a potential source of accidents unless adequate precautions are taken.

Since sparking may even occur with a faultless installation, the more so is the case if short circuits, overloading, faulty installations of the cables, jamming of contacts, fracturing of rope, etc., occur. At this stage only one of such faults is mentioned, but this occurs rather frequently; if, namely, the cross section of the connecting line is insufficiently dimensioned then the large voltage drop will reduce the contacting pressure and as a result violent sparking may occur.

Altogether different is the case of sparks of mechanical origin. Such sparks may be the result of sudden breakdown, e.g., gripping of the brake shoes, fracturing of the rope, etc. Such mechanical breakdowns, however, are very rare, and it is for this reason that they are usually neglected when planning the installation. Furthermore, such sparks are usually of a much lower temperature and duration than the electrically generated sparks.

Explosion Caused by Over-heating.

Extensive experiments were carried out in laboratories to determine the minimum temperature which causes the explosion of various explosive mixtures. It was found that the temperature lies between 150° C. and 750° C., that is between such limits which might easily occur with lifts owing to a short circuit or other mechanical breakdown. A loose connection at a terminal contact or a loose screw might be the reason for such overheating owing to the increased resistance of the circuit at these points.

Reluctant operation of a 3-phase magnetic brake as an example, will result in currents flowing through the windings many times the ordinary working current. As a result the coils burn out, not being dimensioned for such heavy currents, and the heat generated is far in excess of that which would already cause explosion. In D.C. circuits the current is independent of the air gap between magnet and armature, and therefore, overheating will not be caused by such a breakdown. On the other hand, the coils are dimensioned for a short period of operation only, and therefore, a breakdown of the switching apparatus may also in this case bring about extensive overheating.

The windings of the motor armature are also a frequent source of overheating. It is customary to choose a motor of such dimensions as will supply the anticipated power without overheating, but, of course, conditions may be different when the lift is kept in operation for a longer period without rest. With ordinary installations, that is, where the danger of explosion does not exist, this selection of a motor of smaller dimensions than would suffice if very heavy strain is laid upon the lift is justified by considering that there is a rest, by necessity, at each stop while loading and unloading takes place and during which time the motor is allowed to cool off. However, in practice, the operation of lifts varies considerably, and it might happen that in some instances a number of long hoists is required, following in quick succession, thereby causing sericus overheating in the windings.

Apart from the duration of operation, the number of times the contacts make, i.e., the number of movements of the lift is also an important factor.

With 3 phase motors especially the starting current, may be some 6 or 8 times that of the normal current. A large number of short travels, therefore, might be also responsible for dangerous overheating of the armature windings.

With 3 phase motors the failure of one phase might lead to burning out of the windings. It is true that a well designed motor switch will cut out the current in such instances, nevertheless, there are cases when the overload trip fails. Although it would not be pertinent to discuss in this article the reason for such a failure of the trip it might be pointed out that it is due to differential cooling of the motor and trip as well as to their differential location.

As mentioned before overheating due to purely mechanical troubles is not likely to occur, though, for instance, insufficient grease on the guide beams may be the cause of high temperatures. But such faults are just as rare as mechanical breakdowns leading to sparking and the chief object of the designer is still protection against sparking and over heating owing to electrical faults.

Protection against Explosion.

Three customary methods are available for full protection against explosions: (1) Placing the whole of the electrical machinery outside the building where there is no danger of explosion; (2) Completely enclosing the electrical machinery by closely fitting doors, gates, etc., and (3) using specially designed machinery.

Electrical machinery located outside the danger area: Fig. 1 shows an installation for an oil and petrol storing station where the lift is used to convey the oil drums from the basement to ground level. As here we have to deal with a highly explosive mixture of relatively high density, the greatest concentration of the gases will occur in the basement and less only above the ground level. Nevertheless, it would not be safe to place the electrical machinery on top of the building. The simplest solution is, therefore, to do away with all the electrical machinery in the cage and adopt a purely mechanical control for starting stopping of the lift, closing the doors, etc. The hoisting ropes pass through the walls on top of the shaft, and it is most unlikely that explosive mixtures will penetrate through these and get near the machinery (see Fig. 1).

Completely Enclosed Lift and Electrical Machinery: The sclution for this design is shown in Fig. 2, by using "sluice gates" between the shaft and the building. The sluice gates must be airtight and so designed that both doors, the cage door and sluice door, cannot be opened simultaneously. This is achieved, as shown in Fig. 2, by using a double lever arm (a) and (b) 90° displaced to one another and connected rigidly by a rod (c). When (b) is in a horizontal position the sluice door can be opened, at the same time the vertical

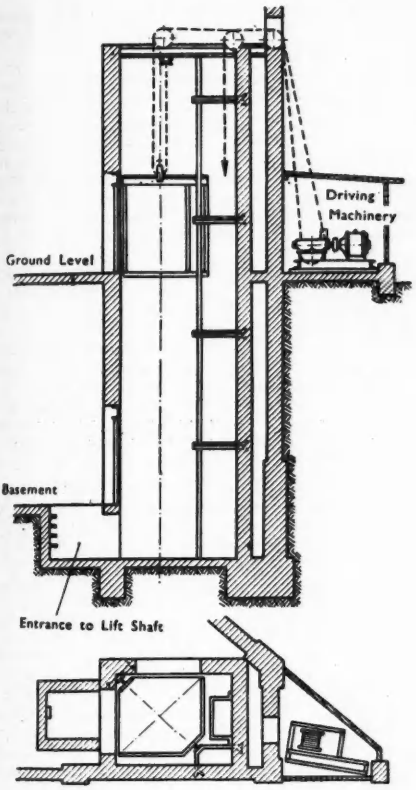


Fig. 1

position of (a) prevents the opening of the cage door and vice versa. r_1 and r_2 are antifriction rollers. A similar arrangement is fitted between the building and engine house.

This solution, however, implies a space limitation just in front of the lift entrance and for this reason it is still customary to do away with the sluice gates and consequently with all electrical machinery in the shaft. To prevent the gases from coming into contact with the motor through the holes for the hoisting rope the motor should be placed as shown in Figs. 3 and 4, depending upon whether the gas is heavier or lighter than air. A still better solution is to place the driving machinery above the top of the shaft as shown in Fig. 5. As can be seen a narrow passage, well ventilated, is arranged between the shaft and motor housing in order to reduce the amount of gas that might eventually enter through the holes for the hoisting rope. It is distinctly advantageous to have adequate ventilation for the lift shaft as well, and if necessary, a forced ventilation should be installed.

Special Protection of Electrical Machinery from coming in Contact with the Explosive Atmosphere : The two previously discussed methods for safety against explosion were based on structural (architectural) modification of the installation. If this is not practicable the problem must be approached from another angle, namely by preventing sparking and overheating of parts which are in direct contact with the explosive atmosphere. Three methods are generally utilized : (1) Enclosing of dangerous parts of the machinery ; (2) Oil immersion of such parts ; and (3) Design with increased safety.

The first of these methods consists of a sufficiently strong casing completely surrounding the machinery where sparks may occur so that explosion inside the casing is not transmitted to the ambient atmosphere.

The bearing surfaces of these casings must be carefully dimensioned for width, length and tolerance to prevent flames and sparks from penetrating to the atmosphere and to cool down sufficiently the hot gases forced out. There are specifications for the dimensions of these casings varying with the nature of the explosive gases in the building. A completely air tight casing would not be practicable owing to the temperature changes of the atmosphere and working conditions. The bearing surfaces must be carefully finished, no varnish or paint must be applied, and they must be slightly greased. Packings are to be avoided.

The second method consists of oil immersion of the dangerous machine parts, liable to spark or overheat. Indicators must be provided to show the oil level in the tanks or casings and must be carefully checked from time to time. The oil bath must be of sufficient depth to prevent the sparks from penetrating through the bath to the atmosphere.

Finally, the method of designing with increased safety is only applicable to cases where sparks, flames,

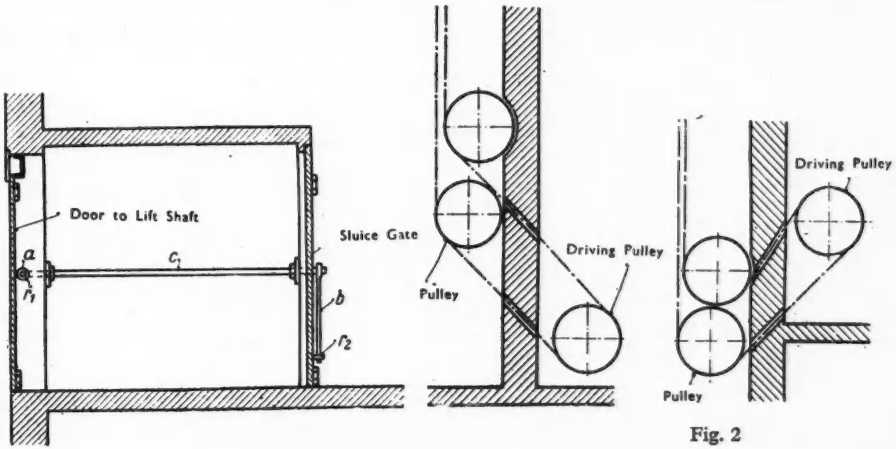


Fig. 2

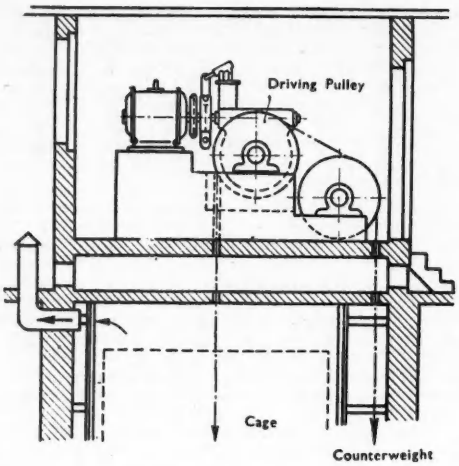


Fig. 3

and high temperatures do not occur in the ordinary manner, and by the specially designed features it is assured that such will not occur, even under extraordinary conditions. It would not be possible to discuss all the safety measures that are customary within the scope of this article and in the following, therefore, only a few points will be mentioned.

Hoisting Motor. Owing to the lack of special specifications ordinary motors enclosed in a casing are generally used, the only additional precaution being that the motor is of such size that even under heavy and continuous loading it will not overheat. It is clear, however, that this is not really a satisfactory protection against explosion and for this reason some manufacturers refuse to quote safe lifts for dangerous buildings.

For full safety it seems advisable to adopt some sort of bimetal operated safety relay in the motor windings, which will automatically interrupt the current when a limiting temperature is reached and re-starting will be made possible only when the motor has cooled down sufficiently. Even this solution is inadequate by itself, as it cannot prevent local overheating owing to some breakdown, and, therefore, it is essential to use in addition an enclosure for the motor.

Switch and Control Apparatus.

For these usually a pressure resistant casing or oil immersion is utilized; D.C. circuits however, do not permit the latter. Protector switches with relays must be supplied together with curves showing the time lag in disrupting the current in order to ascertain that the permissible time to prevent overheating is not exceeded. Fuses are oil immersed or are enclosed and the changing of fuses should be only possible when the power supply is cut off.

For the master switch the motor protecting switch is generally used and is immersed in an oil bath; if a separate master switch is provided, this is usually totally enclosed.

Emergency switches may be either oil immersed or totally enclosed; reversing starter and gear together with auxiliary terminals, press button controls, relays, door contacts, and similar devices are usually in an oil bath. The magnets of the relays must be dimensioned for continuous operation to prevent high temperatures in them.

Cables. Sparking of cables and conductors are mainly due to mechanical damage or short circuits. Excessive heating of conductors can be adequately prevented by using fuses. The choice of the conductors is, therefore, a matter of designing for the anticipated load they have to carry. Mechanical loading of the cables occurs only in the main cables connected to the lift switch board and passing through the shaft. All other cables are prevented from attack by chemicals, gases and water vapour. For this reason conductors are usually of the heavy cable type, with lead or thick rubber insulation. Specifications for full safety of cables are given by the V.D.E. regulations in Germany.

Naturally, great care must be exercised when laying the cables to ensure that they are not stressed mechanically and cable connections should be always totally enclosed.

Summarising, electrically operated lifts in dangerous buildings offer considerable difficulties at the present which can be solved satisfactorily only by architectural methods. If this is not possible the most careful design is essential and it is for this reason that in many instances hydraulically operated lifts are still preferred.

THE USE OF PRESSED METAL POWDERS

By F. CERVINKA. (From *Elektrotechnicky Obzor*, Prague, Vol. 30, No. 3, January 17th, 1941, p. 34).

METALS with a high melting point are easier to form by pressing and sintering than by casting. The original use of this method was for forming pure metals, but nowadays it is used for alloys containing metals having a high melting point as well as for metals with a low melting-point when certain qualities are required, and for non-metallic materials.

Amongst metals platinum and tungsten powder are chiefly used, strongly compressed and then sintered, sometimes under pressure. To ensure good thermal and electric conductivities, silver or copper powder is added to the tungsten powder. Contacts for relays, regulators, contactors, are manufactured in this way.

Another important product filament wire for incandescent lamps, manufactured from a powder containing chiefly tungsten, and additions of tungsten, and various carbides. The resulting products obtained from true metals and carbides, or from carbides only, are very hard and resistant to high temperatures, therefore, they are excellent for the manufacture of all types of cutting tools, wire-drawing dies, etc.

Porous metals form yet another group. The most important amongst these is lead, used for accumulator plates. These plates manufactured by sintering, have a much larger active surface than cast plates. From metallic powders catalysors, filters, wicks for refrigerators are made, and, in general, all parts which must be porous.

While for battery plates the automatically obtained large active surface is sufficient porous metals must be manufactured by adding an easily melted component, such as stearic acid, which evaporates during the sintering and forms capillaries.

In future, bearings will be probably made from porous metals. Such metals have been used for several years in automobile, aero engines and railway

rolling stock, etc. The Swedish steelworks, Fagersta Bruks A.B., produce them according to their own patents. The Teknisk tidskrift number, 37, 1940, give the following information :

Self-lubricating bearings are made from powders containing either copper or iron as principal components, sometimes aluminium or other metals. If copper is the chief component, the porous metal is some type of bronze ; if the chief component is iron, frequently a little copper or tin or lead is added.

The Swedish firm produces porous bearing shells (probably made directly from iron ore). Another component frequently added to bearing metals is graphite, which makes the pressing possible and facilitates the generation of the cellular structure. The bearing shells are sintered in a neutral atmosphere ; usually this is hydrogen, and thus liquid hydrocarbons are formed, which evaporate and form the pores. Graphite also improves the qualities of the shell surfaces.

The shells are pressed nearly to their finished form and then sintered in a neutral atmosphere, bronze at about 800° C., iron at about 1100° C. It is important to avoid considerable changes of volume, and this is obtained by an adequate composition of the powder. After sintering, the shells are pressed into their final form. Machining is not recommended, as the porous structure may be destroyed on the surface. The shells are impregnated with hot mineral oil under a vacuum ; the composition of the oil depending upon the use of the bearing (e.g., for low temperatures). The impregnation is made after the sintering, before the shell comes into contact with the atmosphere. The following are the advantages claimed for these bearings :

(1) No lubrication required.

(2) While the shaft is at a standstill, there is already an oil film. When carrying the load, the temperature rises, additional oil emerges from the pores and improves the lubrication.

(3) No oil can leak out of the end of bearing. Saving of oil is considerable and neither the raw materials nor the manufactured products are spoiled (important for paper mills, textile factories, laundries, etc.).

(4) Control of lubrication is unnecessary. This is advantageous for domestic electric appliances, sewing machines, agricultural machines, etc.

(5) Bearings for high temperatures and for a great temperature range are easy to produce.

(6) There is only slight wear on the journal.

(7) The amount of lubricant required is diminished by 99.9%.

(8) Compared with ball bearings, they are quieter, have smaller diameter and are cheaper.

Disadvantages :

(1) The bearings cannot be repaired on the spot (they must not be machined).

(2) For higher speeds the safe working pressure decreases rapidly. The maximum pressure of 200 kg/cm² may be used only for a very low r.p.m. or temporary loads.

(3) The bearings are most suited to small journal diameters.

(4) For heavy loading and small journal diameter a complementary lubricating system becomes necessary. But even in this case there is a considerable saving of oil and polluted oil may be used, as it is filtered by the bearing itself.

The following are the safe working loads :

Journal circumferential speed in meters per min.	Safe working pressure in kg per cm²
10	13.5
20	11.5
40	9.5
80	7.2
120	6.2
200	5.2
280	4.6

A Swedish porous iron bearing of 7 mm. inner diameter, 11 mm. outer diameter, 18 mm. length is impregnated with 0.3 g of oil only. At 5000 r.p.m. it is charged with 6.3 kg/cm² ; its lifetime is 500 hours.

SAVING FUEL IN GAS-HEATED FURNACE PLANTS

By WILHELM BESSE. (From *Stahl und Eisen*, Vol. 62, No. 8, February, 1942, pp. 156-160).

WHEN the forging plant of the firm Schoneweiss and Co. (Hagen) has been converted from coke heating to gas heating, a very serious increase of fuel (gas) consumption has resulted. Even considering all the advantageous features of gas heating, the price paid for it was by far too excessive. The reason for this uneconomical operation was due in the first place to the inexperience of the workmen in gas heating technique. For example, it took careful coaching of the workers to make it clear to them that there is no need for using flames 1 m. high. But it must be also remembered that with multi-burner furnaces the correct adjustment of combustion, i.e., regulation of fuel/air mixture requires quite some time and effort. Furthermore no measuring instruments were available at first and neither could they be quickly purchased. Nevertheless there was need of fuel saving and this did seem feasible by carefully controlling the furnace operation.

Fuel Saving by Means of Gas Pressure Regulation.

The gas fuel supplied from a long distance under a pressure of 6-7 atm. was at the beginning admitted to the furnace at 5000 mm. w.h. pressure. Owing, however, to the habit of the foundry men to use excess gas, the pressure was reduced to 3000 mm. w.g. to prevent the use of too rich fuel mixture. As a result a very considerable saving in fuel was attained at once.

The plant itself consisted of a number of small and medium capacity furnaces ; the work comprised aircraft parts and high quality forgings mainly for the transport industry. The weight of articles passing through the furnaces was between 0.2 and 80 kg.

Encouraged by the results, the gas pressure was further reduced first to 2500 mm. w.g. and then to 2000 mms. at which pressure the plant is still operating to-day. Fig. 1 shows the fuel consumption in m³/h for 3 burner sizes in dependance of the gas pressure.

Reducing the pressure below 2000 mm. w.g. proved to be ineffective, partly because no further saving could be achieved this way and partly because at lower pressures the carbonization of valves and burners was excessive for reliable operation ; also at lower pressures the operation of the individual burners was mutually affected.

Gauging Gas Consumption.

After the change over to gas fuel, U gauges were installed to each unit to check consumption figures. These simple and cheap instruments proved to be of sufficient accuracy for the job and could be fabricated in the workshops of the plant. So far they have operated for nearly 2 and a half years without failure.

(*To be continued*)

stripped water

In preparing flax and hemp for ropes it is first stripped from its stem, slit into lengths and scraped through a bamboo instrument until only clean fibre remains. This extensive preparation of stripping, slitting and scraping is essential to produce a first-class quality fibre.

The stripping of water of its scale-forming and soluble salts for use as boiler feed is effected by the Permutit "Deminrolit" Process which, not only obtains the results without the use of heat or steam, but also eliminates ammonia and corrosion of condensers.

The modern method of feed water treatment and conditioning is described in our Technical Publication "Distilled Water without Distillation" which will be forwarded on application to The Permutit Company Limited, Dept. T.B., Gunnersbury Avenue, London, W.4. Chiswick 6431.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- PERSONAL
- NOTES

NEW EQUIPMENT

NEW COMBINATION SHARPENING MACHINE FOR REAMERS, HOBS, AND CUTTERS.

It is a remarkable testimonial to the efficiency of the British Machine Tool Industry that, in these difficult times when every section of engineering is working to capacity, designers of machine tools can develop new machines and improve existing types, thus reducing labour in general engineering.

A combination reamer, hob, and cutter sharpening machine which has already been installed in various plants in this country has very interesting features. With this one machine a wide variety of standard sharpening operations can be performed; in addition, several specialized grinding operations can be handled with great speed and economy without sacrifice of accuracy. The sharpening factors are under positive mechanical control, and all mechanical movements of the machine can be positively duplicated to assure uniformity of work on any number of pieces.

The makers of this machine are Barber and Colman Ltd., Brookland, Manchester, and in regard to operating it, the average mechanic can work the machine satisfactorily while carrying on other duties, such as sharpening lathe tools, drills, etc.

HYDRATROL LATHE.

Fig. 1 shows the front view of a Hydratrol lathe, the various speed changes of which are operated—as the name of the machine suggests—hydraulically. The sixteen forward speeds and eight reverse speeds are effected as desired by turning a three-lever handle on front of the headstock. This may be turned without intermediate stop to any speed desired without disengaging the friction driving clutch when a speed change is made. Co-ordinating with the movement of the handle is the Automatic Slide Rule (Fig. 2) which shows spindle revolutions per minute and indicates a slide rule calculation of the cutting speeds in ft./min. Three hydraulically operated friction clutches are carried on the pulley drive shaft. One of these clutches provides eight reverse speeds, and the other two the initial speed progression for the sixteen forward speeds. Spindle release for chucking is obtained by turning a small handle at the front end of the headstock and it can be performed equally well with the spindle running or stopped.

Further details can be obtained from The Broadway Engineering Co. Ltd., Carlisle Road, Hendon, London, N.W.10, who is representing the makers, The Lehman Machine Co.

BONDERIZING FOR TIN CANS AND CAPS.

The Parker Rust-Proof Co., Detroit, has developed a new process for the replacement of tin in the treatment of steel sheets from which thin metal containers and closures are made. Instead of dipping or spraying, the steel sheets pass through all operations, Bonderizing (with the new compound, called Bonderite K), rinsing and drying, on a series of rubber rollers with the dual capacity of conveyors and applicators. Handling is entirely automatic from feeding to packing the finished sheet.

The new product, as applied, produces a fine-grade phosphate coating, providing a rust-inhibiting steel black plate, which when lacquered or enamelled, is successfully used in bending, drawing, crimping, lock-seaming, and other forming operations, without serious loss of finish, adhesion or effectiveness.

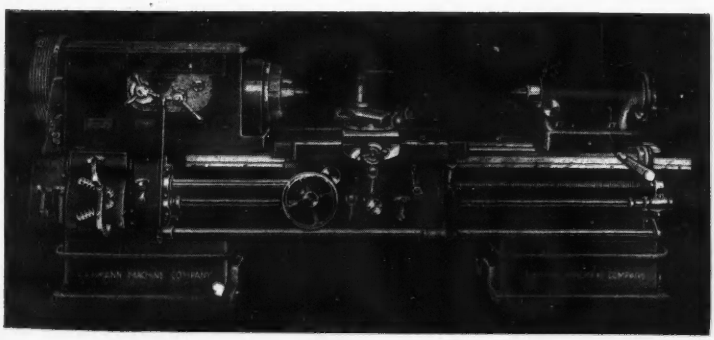


Fig. 1

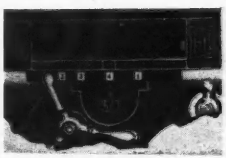


Fig. 2

TEMPERATURE INDICATING STICKS.

UNDER the name Tempilstik°, a new temperature indicator has been marketed by the Tempil Corporation, New York. These new temperature indicators have a specified melting point and are available from 125 deg. F. to 1600 deg. F., in convenient temperature intervals, and are said to have a mean accuracy of within 1 per cent of the temperature stamped on each stick or pellet. For example, a 300 Tempilstik° drawn across a surface heated to less than 300 deg. F. will leave a chalk-like mark, which melts sharply into a liquid streak when the surface reaches 300 deg. F.

This method of temperature determination provides a simple and convenient method of signalling desired preheating temperatures in the welding of steel, special alloys, cast iron and non-ferrous metals; in local heating, heat treatment and fabrication of metals generally, which require careful and accurate temperature control.

CATALOGUES RECEIVED

Broo-Zinc Sand and Diecasting Alloy. A booklet issued by the Brookside Metal Co. Ltd., Astra Works, Harland Estate, Honeypot Lane, Stanmore, Middx., contains interesting data of a comparatively new alloy called Broo-Zinc which has widely been used as excellent substitute for many aluminium alloys. It is about 40% lighter than Brass or Zinc base and has many applications where weight is a governing factor. It has very good casting properties and can be used with equal success in sand, gravity or pressure die-casting. Free copy of the booklet will be sent upon request.

Hancock Oxygen Cutting Machines. A 68-page handbook issued by Hancock & Co. (Engineers) Ltd., Progress Way, Croydon, gives complete instructions for the installation and operation of the Hancock oxygen cutting machines together with hints on how to deal with various jobs and get the best out of the machines at all times. The handbook contains a great amount of useful information for the welder, commencing with data on oxygen and gas suppliers, intensifiers, description of the welding and cutting processes, of burners, tracers, machines, etc. Copies of this interesting and instructive handbook can be obtained on application.

Diesel Electric Locomotives.—For large inducrial plants, mines, and main line hauling, Diesel-electric locomotives of from 10 to 80 tons are described in a new 100-page booklet published by Westinghouse Electric and Manufacturing Co.

Charateristics of Diesel-electric units are compared to steam and mechanical locomotives from the standpoint of availability, reliability, safety, and pulling power. Performance operation, motor and control data are given for 76 typical installations. A discussion of main generators includes construction, insulation, engine starting, and battery charging. Determination of locomotive weight is quickly made from an application chart based on grades and load sizes.

New Air Heater for Ovens.—For heating ovens or furnaces used in annealing of aluminium, glass, and other materials, a new heater has been developed by Westinghouse Electric and Manufacturing Co. Designed for temperatures of from 750 to 1150 deg. F., the unit has a heating element consisting of one piece nickel-chrome ribbon, spirally wound around heavy porcelain insulators. Rating is 5 kilowatts for operation on 220-volt circuits. The complete heater is only 2½ inches thick and requires a space 12 by 33 inches for mounting.

Possible Alternates for Nickel, Chromium, and Chromium-Nickel Alloy Steels.—Four completely new series of alloy steels were announced by the American Iron and Steel Institute. In making the new steels, producers will use less virgin chromium and nickel, and more of these alloys in the form of scrap recovered from steel previously produced. Details of the new steels are given in a booklet prepared at the request of the Office of Production Management, which has been succeeded by the War Production Board.

To develop the new steels, a group of metallurgists chosen from technical committees of the A.I.S.I., the Society of Automotive Engineers, and from alloy manufacturing companies, devised hitherto untried chemical compositions, melted experimental heats of the new steels, tested them thoroughly and selected those which seemed suitable. The alternate steels, presented in the booklet, embrace a series of carbon-molybdenum steels, manganese-molybdenum steels, and also low chromium-molybdenum and low nickel-chromium-molybdenum steels.

The use of the alternate steels, the booklet states, may make necessary some changes in established methods of fabrication or heat treatment, or both, or may even make necessary some changes in engineering design of the product affected.

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